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TRIPLE-EXPANSION ENGINES

AND

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ENGINE-TRIALS.

BY

PROF. OSBORNE REYNOLDS, LL.D., F.R.S.,
M. INST. C. E.

With an Abstract of the Discussion upon the Paper.

EDITED BY
F. E. IDELL, M. E.

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1890.

PREFACE.

The expansion of steam in three or more cylinders is now claiming the attention of engineers and manufacturers, and experimental data upon the action of steam in the cylinders and jackets of a triple-expansion engine will be studied with great care, especially when the engine was designed and built with reference to the facility of measuring each important part while it is performing work.

The Committee, consisting of Prof. Reynolds and Messrs. Ramsbottom and Robinson, having in charge the designing of the engines for the Whitworth Engineering Laboratory, worked with this end in view, and the manufacturer gave them every facility for obtaining an engine which, while representing, as far as possible, the most approved existing practice of steam engine construction, should be thoroughly available for experiment on the use of steam throughout, and if possible beyond the range hitherto attained in practice.

Prof. Reynolds made some preliminary tests and then commenced a series of trials of triple-expansion at 200 lbs. per square inch boiler

pressure with and without steam in the jackets. This series of thirty-two trials was commenced in October, 1888, and finished in April, 1889. Several trials were made at each speed, and the results obtained agreed so closely that three trials with steam in the jackets and three without were taken as illustrating the results obtained; these results are given in detail in the paper.

The Editor has endeavored to retain the full substance of Prof. Reynolds' paper, omitting only such portions as are of no interest to the American readers.

F. E. IDELL.

NEW YORK, May, 1890.

TRIPLE-EXPANSION ENGINES AND ENGINE TRIALS.

In designing steam-engines to take their place amongst the appliances of an engineering laboratory, at the present stage of the development of these institutions, many considerations present themselves.

The primary purpose of the engines is to afford the students opportunities of practice in making the various measurements involved in steam-engine trials, and to afford them an insight into the action of steam in the engine, as well as of the mechanical actions; also to render them familiar with good examples in steam-engine design.

Another purpose, however, which it is very desirable such engines should serve, is that of supplying a means of research

by which knowledge of the steam-engine may be extended. A systematic and experimental investigation of the steam-engine involves two sets of conditions, which, unless it be in a laboratory, can hardly exist together, namely, the time and attention of the scientific investigator, and the assistance of a considerable number of trained observers. In the engineering laboratory these conditions should exist; the first being supplied by the permanent staff, and the second by the students as their training advances.

Having regard to these two purposes, the Committee, appointed by the Council of Owens College to select, amongst other appliances, the steam-engines best adapted for the special purposes of the laboratory, decided that the engines, while as far as possible representing in their principal members the most approved existing practice in steam-engine construction, should be specially designed to afford the utmost facilities for experiments on the use of steam throughout the entire range, and, if pos-

sible, beyond the limits hitherto accomplished in practice.

As best meeting this demand it was decided to have three engines working on separate brakes. All engines to be of the inverted-cylinder type, with the walls and covers separately jacketed, with steam at boiler-pressure, and so arranged that they could be worked with or without steam in any or all of the jackets. Each engine to work with steam at any pressure up to 200 pounds per square inch, to run at any piston speeds up to 1,000 feet per minute, and to have expansion-gear to cut off from zero up to two-thirds of the stroke. One engine to be supplied with air-pump and surface-condenser, the other two engines to be furnished with alternative exhausts, either into the atmosphere, or into steam-jacketed receivers supplying steam to the next engine, each of the receivers also having an alternative supply of steam direct from the boiler. The boiler to be of the locomotive type, having five square feet of grate, to be set in a hot

chamber with an economizer and alternative chimney and forced draught, on the closed stoke-hold system. The condenser to have 200 square feet of cooling surface. The dimensions of the engines to be somewhat as follow:

Engine.	Diameter of Cylinder.	Stroke.	Diameter of Crank-Shaft.
	Inches.	Inches.	Inches.
No. I (high-pressure)	5	10	2¾
No. II (intermediate)	8	10	2¾
No. III (low-pressure).....	12	15	4
Air-pump on No. III	9	4½	
Feed-pump "	1½	2	

In addition to the brake, each engine was to be furnished with a fly-wheel, to act as a belt or rope-pulley, weighing about 1,200 pounds, carried on a separate shaft with a coupling to the crank-shaft.

The design of the engines contains many novelties. These were not adopted without what appeared to the Committee to be sufficient reason, as it was

unanimously desired to adhere as far as possible to ordinary types.

As regards the cylinders, pistons, and valves, there are three noticeable departures; these were adopted with a view—

1. To insure the completeness and efficiency of the steam-jackets.

2. To diminish the resistance to the passage of steam as much as possible.

3. To keep down the clearance.

4. To obtain an adjustable cut-off from zero at any speed.

1. To obtain completeness in jacketing, both ends (or covers) were jacketed as well as the walls. To insure efficiency of the jackets steel liners were used and the covers were domed, so that the surfaces should free themselves by gravitation from the water resulting from condensation, the water being drained from the lowest point in the jacket spaces.

2. To diminish the resistance of the passages, these were abnormally large, the area of the ports being 13 per cent. the area of the piston, and the steam-chests were very large.

3. To diminish clearance, the ports were made straight, and the valves brought as close as possible to the cylinder, double valves being used. The pistons were formed to occupy the space in the cylinder, except one-eighth inch clearance at the ends. The result is that in engine No. I the clearance space shut in by the main valve is 4 per cent. and 1.7 per cent. more by the rider, and in engines II and III the clearances shut in by the main valve are 6 per cent. and 2.5 per cent. more by the riders.

4. To obtain an adjustable cut-off, since at the higher speed the engines were intended to run 400 revolutions per minute, it was practically impossible to use any form of trip cut-off. Meyer expansion-valves were used on the backs of the main valves.

The engines are exceptionally strong, being all of them designed to work safely with a pressure of 200 pounds on the square inch, so that the effect of expansion in one cylinder might be compared with compound or triple expansion.

The frames of the engines are of a somewhat novel form, and their purpose may not be immediately apparent. It will be seen, however, that the front cover is cast with a kind of entablature or box, connected with the base-plate by four wrought-iron columns placed symmetrically as regards the piston-rod. The function of these columns is to withstand the vertical forces arising from the steam pressures on the cylinder covers, and to maintain the axis of the cylinder vertical against any forces; they are not calculated to maintain a horizontal position against lateral forces such as might arise from the action of the slide-block. To meet such lateral forces the base-plate is prolonged upwards in the form of a strong box standard, the upper portion forming the slide-bars, which at the top encircle the piston-rod and pass within, but not touching the box cast on the cylinder cover. Through the sides of this box are four horizontal set-screws, which grip the top of the standard, and so

transmit any lateral force directly to the standard, as well as admitting of the adjustment necessary to maintain the cylinder co-axial with the slide-bars.

In this way the vertical forces are taken symmetrically, and cause no distortion of the engine. The cylinder is held very rigidly by the four columns, and the horizontal forces arising from the pressures of steam in the pipe, and particularly from the expansion and contraction of the pipes under a variation of temperature of more than 300°, are taken by the cast-iron standard. And, what led more than anything else to this design, all distortion arising from heat is avoided. The heat-connection between the cylinder cover at 400° is cut, except for the four columns which are heated symmetrically and the four set-pins which conduct very little heat to the slide-bars.

The result appears very satisfactory, the engines running with the slide-bars cool at 400 revolutions per minute, doing 100 H. P. with great steadiness.

The somewhat peculiar general ar-

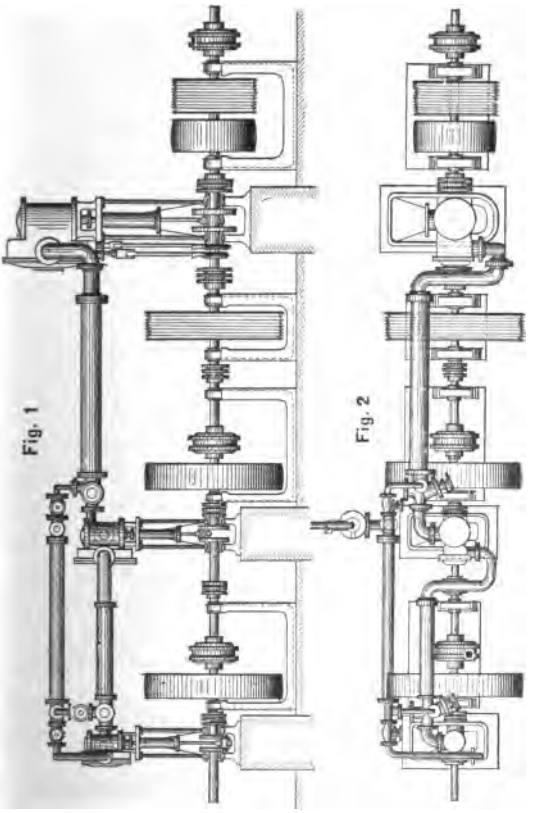
Fig. 1
Fig. 2

rangement of the engines (Figs. 1 and 2) seems to require a word of explanation. Vertical engines were adopted on account of the much greater accessibility they afford to all the parts; also because they allow of the water from the steam-jackets being drained back into the boiler with a less difference of level between the floors of the boiler-house and the engine-room.

The crank-shafts of the engines were raised 3 feet above the floor in order to allow of the floor being kept level and to admit of pulleys 5 feet in diameter. The most noticeable feature in the arrangement of the engines—the distance between them—was necessitated by the alternative shaft connections which it was decided to give them, and particularly by the room required for the belt and rope-gearing, and for working the three separate brakes.

The complete shaft consists of seven separate shafts on separate bearings, which can be connected into a single shaft by six special coupling-boxes. The

shaft immediately on the right of each engine carries a brake, and these brake-shafts of the two smaller engines carry 11-inch belt-pulleys, 5 feet in diameter, weighing 11 cwt., while the brake-shaft for the low-pressure engine carries two 15-inch pulleys, 3 feet in diameter, weighing 9 cwt., one for a belt and one for ropes. These pulleys act as fly-wheels when the engines are working separately; and, in addition to these, there is between the brake-shaft of the intermediate engine and the crank-shaft of the low-pressure engine an intermediate shaft carrying a 12-inch rope-pulley, 5 feet in diameter, weighing 12 cwt., which may be used as an auxiliary fly-wheel on this engine.

When the crank-shafts are working coupled as a single shaft at more than 200 revolutions per minute these larger wheels must be removed from the shafts.

A first-motion shaft, 16 feet distant and 12 feet high, carries pulleys 3 feet in diameter corresponding to those on the engine-shafts, so that the engines can

be geared conjointly or separately on to the first-motion, and this again geared on to one of the brakes, by which means the efficiency of the gearing may well be tested.

The coupling boxes on the main shaft are intended to serve two purposes. (1) To afford a ready means of connecting or disconnecting the several shafts. (2) To allow of any side-play which may arise from the proximity and number of the bearings.

To serve these purposes it was necessary to have a special flexible coupling, which led to the design of a modified form of Oldham's coupling, with an intermediate disk, to which the flanges on the shafts are separately connected, each with two parallel drag-links at equal distances on each side of the shaft. The drag-links, which connect one shaft with the disk, being at right-angles to those which connect the disk to the other shaft, so that the shafts are perfectly free to play laterally. The links are held by pins screwed into the flanges and disk.

To disconnect the shafts all that is necessary is to remove four of these screws and the two links they hold, which leaves the shafts free with a considerable interval between them. These couplings, while very flexible, transmit a perfectly uniform motion and throw no forces on to the bearings.

The intervals between the engines necessitated by this intermediate gearing are 7 feet between No. I and No. II, and 12 feet between No. II and No. III. These intervals entail no evils in the working of the shaft except the increased friction arising from the additional weight and number of the bearings. This friction may be accurately measured and taken into account in determining the brake H. P.

The Arrangement of the Intermediate Steam Connections (Figs. 1 and 2). This was adopted in order—

(1.) To allow of the engines—

Nos. I, II and III being worked as a triple-expansion condensing engine.

Nos. II and III being worked as a compound condensing engine.

" I and II being worked as a compound non-condensing engine.

" III being worked as a single condensing engine.

" I or II being worked as a single non-condensing engine.

(2.) To secure that the steam-supply to each engine, under whatever circumstances it might be working, should be dry without intermediate drainage, so that the weight of water discharged by the air-pump might measure the steam admitted to each engine as steam.

(3.) To bridge over the intervals between the engines without allowing the changes of temperature to cause undue stresses in the pipes and the supports of the engines.

The exhaust-passages from No. I and No. II engines are closed respectively by a 4-inch and a 6-inch steam-valve, while an alternative exhaust-passage, which may be connected directly with an exhaust-pipe in the floor or closed by a

blank flange, is provided. The steam-valves in the exhaust-passages open into receivers which supply steam to No. II and No. III engines respectively, which receivers also have alternative connections with the main steam-pipe, so that each engine can have a separate steam-supply.

The jacketed receivers, which are the intermediate steam passages between the engines, are cast-iron pipes 6 and 8 feet long respectively, lined with wrought-iron pipes 4 inches and 6 inches in diameter, the space between the pipe and casting constituting the space for the steam at boiler-pressure. These receiver-pipes are connected with the engines which they supply by S copper pipes of 4 inches and 6 inches diameter respectively, the copper pipes serving as expansion-joints; the expansion in the 12-foot interval between No. II and No. III engines amounting, with 200 lbs. of steam in the jackets, to 0.25 inch.

The arrangement of the steam-pipe which supplies the receivers was adopted in order that the steam might be dry. This

pipe leads from a water-separator, as a 2½-inch pipe which enters a jacketed receiver No. I, 4 feet long lined with a 2½-inch wrought-iron pipe, to prevent condensation of the steam after leaving the separator. The receiver leads to a point near No. I engine, and is connected with a casting in which are two steam-valves opening into 2-inch copper pipes which lead to the steam-chest of No. I and the receiver between No. I and No. II. The other end of the receiver is connected through a steam-valve with the receiver between No. II and No. III. In this way, whichever engine is receiving steam from the boiler, the steam has to traverse a steam-jacketed receiver.

The position of the boiler and engines (Fig. 3) was adopted to allow not only of the water from the jackets on the cylinders, steam-chests, and receivers draining back into the boiler, but also to allow of its doing so when the pressure of the steam in the separator was 3 lbs. per square inch below that in the boiler.

To insure this, the level of the water in

the boiler is kept 6 feet below the lowest

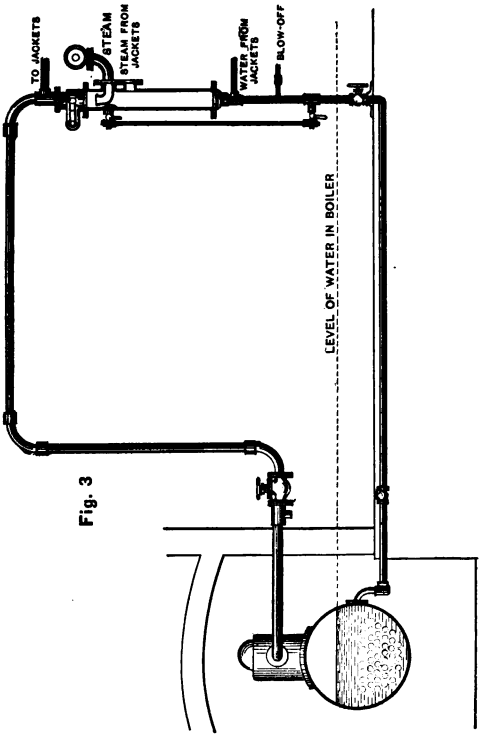

Fig. 3

jacket to be drained. The boiler-house has a floor 5 feet below the engine-room,

and the level of water in the boiler is 1 foot above the engine-room floor, the boiler being 20 feet distant horizontally from the engines.

The steam-pipe, 2½ inches in diameter, takes the steam from the top of the dome on the boiler and enters the engine-room 2½ feet above the floor; immediately in the engine-room is a steam-valve; 2 feet from the wall the pipe rises vertically 8 feet, then turns horizontally for 10 feet, and then again turns down vertically until it enters the separator. At a height of 10 feet there is a branch 2 inches in diameter, without a valve, which supplies all the jackets with steam at the pressure of the boiler less the resistance of the pipe, which is always less than ½ lb. on the square inch. The main pipe then enters the water-separator through a reducing-valve which lowers the pressure 2 lbs.; below this reducing-valve is the steam-pipe leading to the receivers, and below this again the steam-drain from the jackets enters the separator, and 3 feet below this the water drains from the

jackets. The separator now descends as a vertical pipe 1½ inch in diameter to the floor, and then proceeds horizontally until it joins the feed-pipe from the economizer just before entering the boiler, having a back valve and a stop-valve, and also a blow-off valve.

The separator for 3 feet at its upper end consists of a vertical cast-iron cylinder 6 inches in diameter; it is then reduced to a 1½ inch pipe. Communicating with the separator at its top, and at a point 1 foot from the engine-room floor, is a water-gauge of ordinary construction except that the tube is 6 feet long. This gauge shows the level of the water in the separator. When the engines are standing with the blow-off shut, the water remains at the bottom of the gauge. Any water from the jackets drains back into the boiler. If the blow-off is opened the pressure in the separator falls and the water rises to balance the excess of pressure in the boiler, which is shown by the water-gauge; steam is drawn through the jackets, as it cannot pass the reducing-

valve until the pressure has fallen 2 lbs. below the boiler; in this way the engines are heated.

When the engines are running they draw steam out of the separator below the reducing-valve, and hence all the steam is drawn through the jackets until the resistance in the passages reaches 2 lbs. on the square inch; the water in the gauge shows the level at which it stands in the separator. When the pressure in the separator is 2 lbs. below that of the boiler, the water in the separator stands about 5 feet above the floor, which is just the bottom of the 6-inch cylinder; the water then as it enters the separator gravitates to the boiler. If, however, the stop-valve at the bottom of the separator is closed, the water is collected in the 6-inch cylinder, and, as its level is shown on the gauge, this furnishes a ready means of measuring the condensation from jackets and radiation, which measurements may be checked by draining off the water through the blow-off.

In this way the total condensation from

jackets and radiation is determined, and, on consideration, it will appear that herein is an exact measure of all the heat supplied from the boiler over and above that which leaves the engines as steam. It will also be seen that the separator ensures complete water drainage of the jackets and a draught of steam through the jackets and jacket-pipes.

The arrangement of jacket-pipes and drains, which is very complex, was necessary in order that the walls, back and front covers, steam-chest covers, and receiver-covers for each engine might be separately jacketed, and drained both of steam and water. In all there are fifteen separate jackets.

To ensure an equal passage of steam through all these jackets, it would have been desirable, had it been practicable, to supply them in series, so that the steam should pass from one to the other; but this, for obvious reasons, was impracticable, and it was necessary to so arrange the pipes that the head of steam to cause circulation through each jacket should be nearly equal.

This is accomplished by carrying the distributing-pipe, 1½ inch in diameter, throughout the entire length of the engines, as high as practicable. Also the steam-collecting drain, 1½ inch in diameter, and the water-collecting drain, 1 inch in diameter, and arranging them so that there might be a fall all the way in the direction in which the steam was moving. A branch from the steam-pipe with a valve supplies each receiver-jacket on the top, and a drain from the bottom of each receiver-jacket branches into two, one branch falling to the water-drain, and the other rising to the steam-drain, these branches being ¾ inch and ½ inch in diameter.

Each engine has a branch from the distributing-pipe and from each of the drains, which can be closed by valves. The branches from the two drains unite into one drain before branching to the jackets. Then from the distributing branch on each engine are four branches leading respectively to the four jackets on the engines, and in the same way four

drains from the four jackets unite in the one branch from the drain. The jacket-pipes are of copper with iron screwed joints, except the unions, valves, and flanged-joints to the covers, which are of brass. The system is extremely complex, but nothing short of this would suffice for the special purpose of these engines. There are twelve steam-valves, thirty flange connections, and more than forty unions, and about one hundred elbows, tees, and running-joints.

The Covering of Cylinders, &c.—The temperature of the steam-jackets, about 400° Fahrenheit, rendered the covering of the steam-pipes and cylinders a matter of first importance, not only to prevent loss of heat by radiation, but to render it possible to operate near the engines. In the first instance, the cylinders and receivers were surrounded with 2 inches of glass-wool, and lagged with 2 inches of bay-wood, but the glass-wool, being found to create gritty dust, was removed, and an inner lagging of soft pine substituted for it. The steam-chest covers and the

water-separator were also lagged in the same way; while all the steam-pipes, except the copper expansion-pipes and jacket-pipes, which could not be brought under cover of the wood lagging, have been covered with 2 inches asbestos cement.

The surface-condenser is of the torpedo-boat type of thin copper, 14 inches in diameter and 4 feet long It has about 160 square feet of heating surface, and receives the steam by an 8-inch exhaust-pipe from the 12-inch engine.

The air-pump, working by side levers from the slide-block of the 12-inch engine, is 9 inches in diameter, with a 4½-inch stroke, with foot-valve, piston-valve and cover-valve, and is designed to work up to 400 revolutions per minute.

The condenser and air-pump are conveniently placed on a bracket on the standard of the 12-inch engine, which also forms a stage for indicating the engine.

The Feed-pump.—This was adopted in order to maintain a regular feed in the

boiler, as well as to enable the water from the hot-well to be returned to the boiler. It is worked from the rocking-shaft of the air-pump levers; it has a plunger 1½-inch in diameter with a 2-inch stroke, and draws water from a feed-tank 3 feet below it, discharging into a feed-pipe, which, together with the economizer or water-heater, leads through 70 feet of 1¼ inch pipe to the boiler. The inertia of this column of water becomes very considerable when the speed is as great as 400 revolutions per minute, and this, together with the 200 pounds pressure, seemed to render it doubtful whether the pump would answer. However, by means of a special device, a cushion of air or steam was provided about 4 feet from the pump, and by another device the pump was made to start itself, notwithstanding the 3 feet draw, so that the pump works silently and without trouble up to 400 revolutions.

The Governors.—For the special investigations into the action of steam, governors were unnecessary. The load

on the engines being constant, the cut-offs fixed, and the supply of steam regular, small variations of speed would be of no moment; while any alteration of the pressures of steam or cut-off by the governors would only confuse the trials; besides which, the problem of governing engines working in conjunction as regards steam, but on separate brakes, was altogether a new one. At the same time the speed under which the engines were designed to work, rendered it imperative that the engines should be so far governed, that under no circumstances could the speeds exceed a safe limit, which, with the five-foot cast-iron fly-wheels on the shafts, would be about 600 revolutions per minute.

To meet both these considerations, what seemed to be necessary was a safety-governor, which, while it would interfere in no way with the passage of steam at speeds below the limit, would with the utmost certainty cut off steam at some definite speed before the limit was reached.

To ensure certainty of action, it was necessary that the governor should be permanently geared to the engine, and not merely engaged by a belt. And to secure rapidity of action when once the limit of speed was reached, it was desirable that there should be as little room as possible for steam between the governing-valve and the piston; in other words, that the governor should close the expansion-valve.

The Meyer expansion-valves, which had been selected as peculiarly suitable for the purposes of these engines, actuated as they are by screws of such moderate pitch that it requires five or six turns to close the valves, are not susceptible of being opened and closed by the direct force of governor-balls. It therefore became necessary to adopt some form of engagement-governor which, instead of acting on the valve, should act on a clutch which engaged the crank-shaft of the engine with the valve-spindle when the limit of speed was reached. The clutch here adopted is the

Author's spiral steel band-clutch. This clutch, which requires almost an insensible force to engage it, is absolutely certain in its hold.

In order to operate on the valve-spindle it was necessary to use two pair of bevel-wheels, which could not be made less than 4 inches and 6 inches in diameter. To throw this train of wheels suddenly into gear with a shaft making 400 revolutions per minute seemed a doubtful proceeding, but such is the softness of action of the clutch, although there is no slipping, that there is neither noise nor shock. The engagement is silent and instantaneous, so that unless special attention is directed to it the movement of the 10-inch hand-wheel will probably escape notice. The clutch is as good in disengagement as in engagement, and will release the shaft before it has turned more than 5° or 10°.

Although the main object of these governors was that of a safety-governor, opportunity was taken to so design them

that they should, if required, open the valve as the speed fell, as well as close it as it rose, arrangements being made to prevent hunting. The governors so obtained are extremely efficient, and afford an excellent means of studying the action of governors. During the steam trials, however, they are simply set to act as safety-governors, which they have done to perfection, never having been out of action, or having allowed the speed of the engine to exceed the limit to which they are set.

The boiler is of the locomotive type with iron tubes and firebox, the shell being of steel $\frac{9}{16}$ inch thick. The tubes are 2 inches in external diameter and 8 feet long, giving 160 square feet of tube surface. The firebox is $\frac{8}{16}$ inch thick, 2 feet 3 inches by 2 feet 4 inches, 4 feet high, giving 42 square feet of heating-surface.

The area of the grate as used is not more than 4 square feet.

The boiler is furnished with a dome, from the top of which the steam-pipe descends and passes out at the side.

The feed enters the boiler just below the water-level and in front of the fire-box.

There is an iron smoke-box at the end of the boiler, from which there are several passages for the gases. The usual passage is beneath the barrel of the boiler, 3 feet broad and 6 inches deep, and about 6 feet long, proceeding at a slight inclination downwards towards the fire-box; across this passage the feed-pipe ranges backwards and forwards, and a series of scrapers are worked to keep the pipes clean. The pipes cross forty times, and give about 50 square feet of heating-surface, 40 square feet of which is kept clean by the scrapers. In this arrangement the water ascends in the opposite direction to that in which the gases descend. The gases, after emerging from the water-heater, descend into a flue leading to the chimney, which is 100 feet high, and takes the gases from other furnaces, affording generally about $\frac{3}{8}$ inch draught.

The boiler and water-heater are en-

closed in a brick chamber arched over. This chamber is 6 feet wide and 9 feet high, extending from the front of the firebox to the end of the smoke-box.

At the firebox end a second chamber is built 6 feet by 6 feet and 8 feet high. This, by shutting a door, becomes a closed stokehold, into which a fan can be used to force air at any pressure up to 2 inches of water.

In this chamber is an injector, a feed-tank and water-supply, a window looking at the safety-valves, and a window into the engine-room, also a tumbling-hopper for admitting coal.

There are two 1-inch dead-weight safety-valves on the boiler, loaded to 200 pounds, as well as the usual fittings.

The Measuring Appliances.

These, in respect of the brake-dynamometers, the indicating gear, the gauge for jacket-water, and the tumbling-bay and tank for the condensing water, are of a permanent character. Provision is also made for measuring the temperature of

the gases in the smoke-box as they emerge from the tubes, and in the flue as they leave the water-heater, and for measuring the temperature of the feed before passing the pump, and as it enters the boiler after passing the water-heater.

The condensing water is drawn from an iron tank 20 feet by 10 feet by 10 feet, about 116 feet above the engine-room floor. A permanent mercurial gauge in the engine-room always shows the level of water in this tank. The great head, although, of course, a waste of power, is of advantage in securing a regularity of flow. The water after leaving the condenser enters a cast-iron tank, 4 feet by 18 inches by 18 inches, from which it issues over a tumbling-bay 4 inches wide; in the tank are bafflers and a float, with a scale graduated to show in pounds per minute the quantity running over the bay. The water is then caught in a second receiving tank and conducted to an underground concrete tank 20 feet by 9 feet by 11 feet, the level of water in which is shown in the engine-room by a water-gauge, and

also indicated outside by a float. This tank, which has been accurately measured, affords a very exact means of checking the indications of the float in the tumbling-bay.

The upper tank holds 12,000 gallons of water, which can be passed through the condenser before the tank is empty. When the upper tank is empty, if more water is required the quadruple centrifugal pump is set in motion, which raises the water at the rate of 10,000 gallons an hour from the lower to the upper tank; but it is seldom necessary to resort to this. The temperature of the condensing water is measured by a thermometer in the pipe leading to the condenser, and after leaving the condenser by a thermometer in the float-tank.

The water from the hot-well flows into an oil-separating tank, from which it overflows on opening a cock, and is caught in a 100-pound tip-can, from which it may be tipped into the feed-tank, so that the feed and hot-well discharge is measured at one operation.

The condenser is furnished with a mercurial gauge, which shows the absolute pressure in the condenser; also by a Bourdon vacuum-gauge, and the temperature of the discharge from the hot-well is measured by a thermometer in the hot-well. The water, resulting from radiation and jacket condensation, is measured in the water-separator.

The pressures in the receivers are shown by Bourdon gauges, graduated to pounds.

Each engine is provided with a counter for recording the revolutions.

The Indicating Gear.—The indicator cocks have a clear ½-inch way into the cylinder, the cock being placed at the end of a stiff brass tube screwed horizontally into the cylinder, and reaching through the 4 inches of lagging. The cock itself forms an elbow, to allow the indicator to have a vertical position.

The cocks from the back, and from the front of each cylinder are in the same vertical line, so that the indicators stand vertically over each other in a conveni-

ent position to receive the motion for the drums. This is obtained, in the 12-inch engine from the air-pump levers, and in the other engines from levers specially connected by a link with the slide-blocks.

In all cases the indicators are some feet above the levers, and while the motion of the levers is vertical, that of the drums is horizontal.

The Hydraulic Brake Dynamometers.—These are a very important feature of the system. They are the result of a special investigation as to the possibilities afforded by hydraulic brakes, undertaken by the Author during the time when the engines were under the consideration of the Committee, and before anything was decided.

Having had a great deal of experience with almost every conceivable form of friction brake, the Author had arrived at the conclusion that, although it is possible to construct such brakes to work with almost any degree of accuracy, certain inconveniences and drawbacks at-

tend their use, which in all cases leave much to be desired, particularly where, as in a case like this, work on the brake is the sole object of the engines.

(1.) Such brakes require constant observation and watching.

(2.) A single engine cannot be started without relieving the load.

(3.) Such brakes are cumbersome and are not easily adapted to measure greatly different powers.

(4.) Any particular brake cannot without considerable pulling about, such as altogether removing the brake and brake-wheel, be rendered altogether nugatory. It was desirable:—

1. That the brakes should be certain in their action without any attention while the engines were running.

2. That they should leave the engines free to start, and then take up their load without attention.

3. That they should be put on and off by a simple operation.

4. That when turned off they should offer no sensible resistance to the engines.

5. That they should be capable of being so adjusted as to impose any particular resistance, from zero to the greatest, at any speed at which it was desired to run the engines.

6. That the resistance of the brake, when once adjusted, should be independent of the speed of the engine.

7. That the necessary size and structure of the brakes should not be such as to incommode or hamper the engines.

8. That the resistance of the brake should admit of absolute determination from a single observation.

Of these attributes 1 and 2 belong to all fluid resistance, such as that of the screws of steam-ships or centrifugal pumps, in which cases the resistance, varying as the square of the speed, is zero when the engines start.

If the casing of a centrifugal pump, or the tank in which a paddle or screw works, be suspended on the crank-shaft, making a complete balance when the shaft is at rest, then, when the shaft is in motion, the moment of resistance on

the shaft will be exactly equal to the moment to turn the casing round the shaft. This can be readily and absolutely measured by suspending weights at a definite horizontal distance from the shaft. The first published account of this form of brake having been made use of for dynamometric measurement was by Hirn,[1] in his investigation for the verification of Joule's mechanical equivalent of heat, and was subsequently adopted by Joule in his second determination.

In neither of these cases, to the Author's knowledge, was there any attempt to vary the resistance at a constant speed.

Having occasion to use a dynamometer for measuring the resistance on the shaft of a multiple steam-turbine at speeds of 12,000 revolutions per minute, which was engaging his attention in 1876, the Author made use of a brake, having a centrifugal pump suspended on the

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[1] Théorie Mécanique de la Chaleur, 2nd edition, 1865, p. 55.

shaft and working into itself. The resistance, or head against which the pump was working, was regulated by a valve between the exit and inlet passages, that is, in the external circuit made by the water. This was brought before the Mechanical Section of the British Association in 1877. At the same meeting, Mr. William Froude gave an account of his hydraulic brake, for measuring the power of large engines, in which the resistance was regulated on the same principle as that adopted by the Author, namely, by adjusting diaphragms or sluices in the passages between the revolving wheel and the casing. In other respects Mr. Froude's brake differed essentially from any of those previously used, being designed to obtain a maximum resistance with a given sized wheel. For this purpose Mr. Froude invented an internal arrangement which affords a resistance out of all comparison with any other form.

Since great resistance, admitting of small brakes, was of extreme importance

for these engines, the first step in the special investigation was the construction of a model Froude's brake with a 4-inch wheel; the object of which was to ascertain how far the sluices would act in maintaining a constant resistance at any particular speed, and what was the minimum resistance when the sluices were closed.

With this brake it was found that the minimum resistance was about 0.08 of the maximum; a hardly satisfactory range, considering it was desired to run the engines at a constant load at from 100 to 400 revolutions per minute, the maximum resistance of the brake ranging from 1 to 16, so that the minimum at 400 would be 26 per cent. greater than the maximum at 100 revolutions, apart from the fact that closing the sluices would not render the brake nugatory.

This, however, was of small importance compared with another fact revealed by these experiments. When the speed of the brake-wheel exceeded a certain small limit, determined by the head of water

under which it was working, the maximum resistance gradually fell off in a surprising and somewhat irregular manner. This falling off was found to be owing to the brake partially emptying itself of water, due to the air from the water gradually accumulating in the center of the vortex—a fact which, if not dealt with, threatened to render such brakes useless for the purpose of these engines.

The argument was simple: in a vortex the pressure at the center is less than the pressure at the outside. The pressure at the outside in these brakes is determined by the atmosphere, and the small head under which they are working; and the outside forms a closed surface. The pressure at the center will therefore, at different speeds, fall below the pressure of the atmosphere. Air will be drawn from the water and accumulated in the center, occupying the space of the water and diminishing the resistance; and, owing to various causes, the action will be irregular. This would be

prevented if passages could be carried through the outside to the axis of the vortex, carrying a supply of water at or above the pressure of the atmosphere, so to prevent the pressure at this point falling below that of the atmosphere. This was accomplished by perforating the vanes of the wheel, and supplying water through the perforations. It also appeared that, by having similar perforations in the casing open to the atmosphere, the pressure at the center of the vortex could be rendered constant, whatever the supply of water and speed of the wheel; so that it would then be possible to run the brake partially full, and regulate the resistance, from nothing to the maximum, without the sluices. These conclusions having been verified on a model, it was decided to arrange the engines with the shafts in line, with three brakes on the shafts; and the brakes, with 18-inch wheels, were designed according to the resistance given by the model. The brakes promised all the attributes desirable, except that of running

with a constant load under varying speeds. This matter was considered during their construction, and an automatic arrangement was devised acting on cocks regulating the supply and exit of the water to and from the brake necessary to keep it cool, the lifting of the lever opening the exit and closing the supply, so as to diminish the quantity in the brake, and *vice versa.*

The danger of such an arrangement hunting was carefully considered, and precautions were taken. The brakes were constructed at the same time with the engines, and the engines started with the brakes and automatic gear complete. During the twelve months they have been running the brakes have demanded and received no attention whatever. They are easily tested for balance. They have neither fixed nor spring attachment, except the bearing on the shaft. They are loaded on a 4-foot lever, with 2-inch play between the stops. When the speed of the engines reaches about 20 revolutions per minute, the levers rise

(whatever load they have on), and, though always in slight motion, they do not vary ½ inch until the engines stop; during the run the load on the brakes may be altered at will, without any other adjustment.

The Engine Trials.

Before commencing the trials, the object to which they were to be directed, and the manner in which they should be conducted, were carefully considered, and it was decided:—

1. That the purpose of the trials should be the elucidation of the general laws of the action of steam in the steam-engine, and the more general circumstances on which these laws depend.

2. That, from the commencement, the trials should be systematic; certain definite conditions being aimed at, and the trials under each set of conditions continued until consistent results should be obtained, showing how far the conditions had been achieved.

3. That there should be no casual

nor unrecorded trials, but that all trials should be considered of the same degree of importance.

4. That observations should be noted and reduced on special forms according to a definite system, to be carefully preserved for future reference; and that a synopsis of the mean results of each trial should be entered forthwith in a special record for ready comparison.

The trials have all so far been conducted as part of the regular work of the laboratory, under the superintendence of the Author, Mr. Foster (assistant in the laboratory) having general charge of the appliances. The detailed observations were taken and reduced by students (about fourteen on each trial) under the supervision of Mr. Mackinnon, demonstrator of the laboratory.

Diagrams are taken every half-hour simultaneously from the six ends by six students, who have charge of their respective indicators for the trial. The same students also reduce the diagrams in the intervals. The three counters are read

every ten minutes by three students, who have respectively charge of the counters and running of the three engines, calculating the brake H. P. as the trial proceeds, and noting any circumstance connected with the resistance or running of the engine.

One student has charge of the 100-pound tip-can, which measures the water from the hot-well; and another has charge of the condensing water, noting the temperature and quantity given by the float every ten minutes. Another student measures the rate of discharge from the jackets every half-hour. A student watches the coal-weighing and firing. A student takes the temperatures of the hot-well and feed before and after passing the economizer, and the temperature of the air in the smoke-box and flue before and after passing the economizer. Each student reduces his observations as he proceeds, so that within a few minutes of the end of the trial the reduction is completed.

The results are then examined, checked

and entered in the permanent record, the original diagrams and notes of each trial being carefully preserved.

The fire is lighted the first thing in the morning, and steam is got up quickly. As the steam rises it is blown freely through the jackets to heat the engines. If the trial is to be made with jackets, the blowing through all the jackets is continued until the boiler-pressure reaches 200 pounds on the gauges. Should the trial be without jackets, the jacket-covers on the low-pressure engine are closed when the pressure has reached about 40 pounds, and the air-cock is opened; those on the intermediate cylinder when the pressure reaches about 80 pounds, and those on the high-pressure cylinder at 200 pounds. In all cases the engines are started, and are allowed to run just as required for the trial for one hour. The engines are then stopped fifteen minutes before the trial, the fire is drawn, and the readings of the counters and level of the water in the boiler and tanks are taken; 14 pounds of

wood and 14 pounds of coal are allowed for the waste of re-lighting, starting and stopping. The run then commences; the coal is weighed out in charges of 100 pounds, each charge being shot from the scale-pan into the hopper in the firing-chamber, and completely consumed before the next weighing is admitted.

The boiler is fed continuously by the feed-pump, either from the water from the hot well or, in some trials, from the water from the condenser. The runs have generally been for six hours, except when forced draught is used, in which case they are about four hours.

After the last coal has been put upon the fire, the engines are run as long as steam can be kept up, care being taken to bring the level of the water in the boiler at stopping exactly to that at starting, any difference being allowed for as 15 pounds for each $\frac{1}{10}$ inch.

The ashes which fall through the bars are burned during the trial, and the ashes after the trial are generally weighed, but no account is taken of them, nor

of any fuel that may be left in the grate.

This was adopted, after trying several systems, as being workable and very definite; nor does it appear, on comparing the results from the long with those of the short trials, that the one has any sensible advantage over the other. During the experiment the regulator is fully open, and a definite quantity of water run through the condenser. The engines, therefore, take all the steam the boilers will produce, the load on the brakes just balancing the pressure of steam, so that the speed is regulated by the rate at which steam is made in the boiler, that is, by the draught-gauge. As it was intended that the scope of these trials should include as far as possible all conditions under which steam may be used, there was no particular reason for commencing with one set of conditions rather than another, except such as arose from convenience, and out of consideration for the engines themselves. The fact that the engines were new, and

wanted running to bring the bearings into order, as well as the number of students to be employed, led to the first series of trials being made with triple expansion and full pressures of steam.

The Results of the Trials.

The trials commenced in March, 1888, and were continued at the rate of two a week till June; in all twenty trials were made and recorded, the engines being then complete with the exception of lagging.

These early trials with 200 pounds pressure triple expansion, with and without steam-jackets, and various degrees of expansion, gave very definite results. But they also revealed the fact that the linings of the cylinders leaked at pressures above 170 pounds per square inch, and that the joints in the jacket-pipes could not be made to hold. They also showed that, notwithstanding the precautions taken, the jackets were liable to fall off in efficiency. The effect

of the leaks was not great on the general economy of the engines, and might easily have passed unnoticed but for the rigor of the tests to which they were subjected.

At 250 revolutions per minute the thermal efficiency of the engine with jackets was—

$$\frac{\text{Heat equivalent of indicated work per minute}}{\text{Heat discharged + heat equivalent of indicated work}} = 0.175$$

Coal per H. P. per hour . . = 1.48 lb.

The leaks, however, tended to confuse the diagrams, and opportunity was taken of the long vacation, during which the trials were discontinued, to re-set the linings of the cylinders. The lagging of the engines was completed as far as it was thought desirable.

The trials were continued in October, when the linings proved to be perfectly tight, and although at first the jacket-pipes leaked occasionally, the leakage was not of any sensible magnitude. The

jackets were, however, still found liable to fall off in effect at low speeds. The trial with jackets was therefore repeated many times, small alterations being made in the jacket-pipes, until consistent results were obtained with speeds of 250 revolutions per minute, giving thermal efficiency, calculated as before, 0.20, coal per indicated H. P., 1.33 pounds. Corresponding trials without the jackets were then made, followed by trials at higher and lower speeds with and without the jackets. These furnish a complete series of trials of triple expansion engines working with about 200 pounds boiler pressure, at piston speeds from 250 to 1,000 feet per minute.

Table I shows the mean results as recorded for three trials at different speeds with and without jackets. Only one trial at each speed is given, though several trials have been recorded, the results not differing by 1 per cent.

Lines 4 to 29 contain the mean results from the engines.

Lines 30 to 42 contain the heat discharged from the engines.

Lines 43 to 48 contain the heat received by the engines.

Lines 49 to 59 contain the heat received from the furnace.

Lines 60 to 76 contain the general relations between the coal, heat, water and power.

It will be noticed that the three engines do not run at the same speed in the same trial. This is a matter of great importance, and shows the advantage of having for such trials as these the engines working on separate brakes.

The cut-off in each cylinder regulates the fall of pressure in that cylinder, but the pressure in the receiver into which it discharges is determined so as to equalize the steam received, and the steam drains off into the next engine.

If, then, the shafts are coupled, there can be only one ratio of expansion, which will make the terminal pressures in the cylinders correspond with the

pressures in the receivers. But when the shafts are free the engines adjust themselves so that they pass the same quantity of steam, and the cut-offs are easily arranged to bring the terminal pressure into accordance with the pressure in the receivers. Thus, with these three separate engines, the full economic advantage of all degrees of expansion can be obtained. To do this with coupled engines would require a different ratio of cylinder volumes for each degree of expansion, these trials showing distinctly what should be the cylinder volume for each degree with coupled engines.

The Checking of the Results.—The system rendered possible by the use of a surface-condenser, of accurately measuring the water which has passed through the engines, as well as the heat discharged from the condenser, and the feed-water, gives a certainty to the results of the trials not otherwise to be obtained. There will be always a loss between the water supplied to the feed-

pump and that received by the engines; hence, unless the loss is definitely known, the actual water received by the engines can only be surmised.

In the first forty of these trials the water discharged from the engines after being measured has been returned to the boiler, the deficiency being carefully ascertained; and in no case where this has been done has the deficiency amounted to less than ½ pound per minute, although there were no visible or perceivable leaks of any sort from joints or glands, and the boiler, when tested before and after the experiment with water-pressure, has shown no leak. Great pains have been taken to find where this water went, but without success, though it certainly did not go through the engines.

The importance of this point in determining the action of steam in the cylinder is fundamental. It is only by knowing the quantity of water passing through the engines that it is possible to compare the actual diagrams with a theoretical diagram: and the difference be-

tween the feed and the hot-well discharge would in these engines generally amount to from 5 to 10 per cent., and would vitiate any such comparison. As it is, all comparisons have been made from the water discharged from the hot-well.

Since each pound of dry saturated steam condensed would give up about 1,000 thermal units to the condensing water, the measures of water from the hot-well and heat from the condenser keep a useful running check upon each other. It is found that the heat measured (in 1,000 thermal units) is about 4 per cent. greater than the water measured in pounds when the jackets are on, and from 1 to 2 per cent. less when the jackets are off.

An exact calculation, as to the heat discharged per pound of water, must involve certain assumptions, of the accuracy of which a careful comparison with the measured heat affords a valuable test. Such a comparison is shown in Table II.

For the trials with the jackets on, the calculations are made on the assumption

that the steam is released as dry saturated steam, and carries with it into the condenser the heat of evaporation at release pressure from the temperature of the hot-well, less the external work of evaporation and plus the work done by the piston in discharging the exhaust. This, expressed in quantities from Professor Rankine's table, is

$$\frac{H_2 - h_3 - (P_2 - P_3)\ V_2}{772}$$

In this calculation no account is taken of the additional heat received by the steam during its passage from the cylinder into the condenser from the hot walls of the passages.

For the trials without jackets, the calculations are made on the assumption that the steam is admitted into the low-pressure cylinder as dry saturated steam, carrying into the cylinder the total heat of evaporation from the temperature of the condenser at the temperature of admission, and that it carries this heat, less the heat equivalent of the indicated

work done in this cylinder per pound of steam, into the condenser, which, expressed in Professor Rankine's quantities, is—

$$\frac{H_1 - h_2}{772} - \frac{(\text{I.H. P.}) \times 42.7}{\text{lbs. per minute from the hot-well.}}$$

This calculation, therefore, takes no account of the heat that must be lost by the steam in supplying the heat to be radiated from the exterior of the cylinder.

Since important actions are not taken into account in these calculations, the resulting quantities cannot be considered an absolute check upon the observed quantities; they constitute, however, a valuable relative check. Thus in trials 44, 33, 56 (with jackets) the observed discharges of heat are greater than the calculated by amounts which diminish slightly as the speed increases. These differences, about 5 per cent. of the total heat discharged, which will be the subject of further remark, reveal no inconsistency in the observed results, which so far check each other. On the other

hand, in the trials 41, 35, 40 (without jackets), while the observed discharges (for trials 35 and 40) are from 1 to 2 per cent. below those calculated, allowing a margin for external radiation, the observed discharge for trial 41 is about 5 per cent. larger than the calculated, an inconsistency which shows error of observation somewhere. Table II does not supply sufficient evidence to locate the error, but this is found in Table I in the quantities given under the head radiation (line 41).

This radiation is obtained as the balance of the total heat received from the boiler (in the water from the hot-well as dry steam, and in the jacket water), and the total heat discharged as heat and work; hence any error in measuring the heat discharged, or the water from the hot-well, would affect the apparent radiation. Since all the trials without jackets are made under approximately the same radiating conditions, and these conditions are such as would cause slightly less radiation than the trials with jackets,

the actual radiation in the trials without jackets must have been nearly the same, and somewhat less than in the trials with jackets. In Table I the radiation for trial 41 is 503 thermal units per minute, 897 for 35, 1,170 for 40, and 1,266 for the trials with jackets, so that the radiation in trial 41 is clearly some 500 thermal units per minute too small. This might be due to an error either in the hot-well discharge or in the heat discharge; but as the former would affect the heat per lb. of coal (line 62), and so bring this trial out of accord with the others, it seems that the error is in the heat discharged.

The correction that would bring the observed heat discharged in Table I, trial 41, into accord with the others is 60 thermal units per lb., or 460 thermal units per minute, which heat, transferred to the radiation, would bring this to 963, or nearly the mean of that for trials 35 and 40. This shows the completeness of the check throughout these results.

The Radiation.—The slight differ-

ences which are shown in this quantity, Table I, line 41, for all the trials with jackets, may have been due to differences of temperature in the engine-room. The mean radiation with 200 lbs. steam in the jackets is 1,266 thermal units per minute, and the mean radiation in the trials with the cylinder jackets shut off (omitting 41) is 1,037, the difference being 229, with or without jackets, at a pressure of 200 lbs. per square inch. This is exclusive of radiation from the boiler.

The Heat Abstracted during Exhaust.—That during the exhaust the water in the cylinder, which has resulted from condensation, is re-evaporated by heat from the walls is well established, and it has been often suggested that the steam leaving the cylinder may be somewhat superheated by the hot walls of the passages. The excess of the observed heat discharged over that calculated in Table II, might be explained by the second of these causes, but not by the first, since the diagrams all show that

the steam was in the condition of dry saturated steam at release; besides which, the calculated heat takes account of all the heat it could so possess. To account for this difference, which amounts to 5 per cent. of the total heat discharged, by supposing the steam superheated would be to suppose the temperature of the steam raised from 70° to 100° above the temperature of the condenser. Considering that the temperature of the steam in the jackets was 250° higher than that in the condenser, there would be nothing apparently impossible in thus superheating the steam while passing through the ports and exhaust passage heated by the jackets. Such a rise of temperature would, however, be apparent in the exhaust pipe if sought for; and as thermometers showed that the temperature of this did not rise at any time to more than 140° Fahrenheit, which temperature corresponded with the pressure of steam in the condenser, it is evident that this heat did not go to raise the temperature

of the effluent steam. The fact that the difference varies so little with the speed of the engine suggests that this absorption of heat is consequent, in some way, on the mechanical action to which the steam is subject during exhaust, in a similar manner to that in which the heat supplied by the jackets to the cylinder is consequent on the expansion, and this appears to be the case.

The steam in the cylinder at release expands down to the pressure of the condenser. The expansion takes place partly in the cylinder, partly in the passages, and will be attended by liquefaction similar to that which results from ordinary expansion. The liquid, thus formed, may be re-evaporated, from the hot walls of the cylinder and the passages, without raising the temperature of the steam above that of the condenser. This expansion is from the volume (per lb.) at release to the volume (per lb.) at the pressure in the condenser, and the amount of heat for re-evaporation can be definitely estimated. In trials 44, 35, 56

respectively, this heat amounts to 84, 87, 71 thermal units per lb. of steam. Some considerable portion of the heat would be supplied from the work done by the steam against the resistance in the passages, which would be directly reconverted into heat; but the greater portion would have to be obtained from the surfaces, or else the steam would enter the exhaust in a supersaturated condition. The excesses of the observed heat over the calculated, Table II, are 64, 29, 31 thermal units per lb., being well within the heat necessary to re-evaporate the water, after making allowance for the friction of the passages. This heat, it is to be noticed, is acquired by the steam from the walls after the steam has done its work in the cylinder, and must be supplied either by the jackets or by the condensation in the steam-chest, ports, and cylinder. It therefore represents heat which passes direct through the engine, without effecting any work, and is a loss of between 3 and 6 per cent. of the theoretical efficiency of the steam.

The Diagrams have been taken with six Crosby indicators, and with springs as low as the speeds and pressures would admit.

The reduction is effected by measuring ten breadths, the pressure and back-pressure from the atmospheric line, and then the effective pressure, so that the results check, and may be directly used to obtain a mean diagram. These results have been several times checked by a planimeter, without establishing any sensible difference. As regards the diagrams themselves, every precaution has been taken to insure accuracy, and there is no reason to suppose that there are any prevailing errors of 1 per cent., although errors of the instruments, and, indeed, of all indicators, when subjected to certain particular tests, are much greater than this. The check afforded by the brake-power, although it would not reveal a prevailing error of 2 or 3 per cent., has this important effect, that it does away with any possible bias that might result from enthusiasm to obtain high indicated power,

for by so doing the effect would be to lower the mechanical efficiency of the engine.

It is, however, the consistent agreement of the curves of expansion, as indicated, with the theoretical curve for the weight of absolute steam shown by the water discharged to have passed through the engine, that gives the greatest confidence in the indicated results.

The Reduction of the Diagrams to a mean Compound Diagram.—Considering the important place which must be occupied by mean compound diagrams, in comparing the results of the various trials in such an extended investigation of the steam-engine, it was necessary that some system of reduction should be adhered to, and the choice of this system was a matter of the first importance. There was one peculiarity about the working of these engines which necessitated a departure from any methods previously adopted, namely, the unequal speeds of the three engines. This fact had great influence in determining the

system adopted. Except as affected by this, the methods of reduction did not differ from one or other of the plans usually followed.

The reduction of the twenty-four diagrams, taken during a trial from each engine, is effected by finding the means of each of the twenty measured distances from the atmospheric line, which are then reduced to a common scale, 10 lbs. to an inch. These ordinates are then plotted, so as to project the diagram to a length determined, as will be subsequently described. The volumes of clearance, 4 per cent. on engine I, and 6 per cent. on engines II and III, valve-clearance 1.65 per cent. on engine I, and 2.5 on engines II and III are then added to obtain the line of zero volume. Thus, a compound diagram is obtained showing the relation of volumes and pressures of the whole steam in each of the cylinders. To reduce this diagram, to show the relation of volume and of pressure of the steam discharged from the cylinder, an ideal compression-line is drawn through the point of the

actual compression-curve which corresponds to the closing of the exhaust. Horizontal lines are next drawn across the diagram, cutting the expansion-curve, the compression-line, and the ideal line, and each of these horizontal lines is set back until the point which was the ideal compression-curve reaches the line of zero volume. Then the positions taken by the points from the expansion-line and the actual compression-line show the volume of steam in the cylinder over and above the volume of that which is shut in at exhaust. All this reduction may be done arithmetically, or by plotting. The result is that, while the area of the diagram has not been altered, the actual expansion and compression-line for the steam passing through the engine is obtained; Rankine's curve of saturation for the weight of steam discharged is then drawn. On account of the varying difference between the speeds of these engines, the lengths for the compound diagram could not be obtained by simply projecting the lengths of the separate diagrams, so that they

should be proportional to the effective volumes of the several cylinders. It was necessary to project them so that they should be proportional to the products of the effective volumes of each engine multiplied by its revolutions per minute. Slight as this necessary modification may appear, it does away with the idea of a relation between the area of a diagram and the size of the engine, which, once got rid of, leaves it apparent that the separate diagrams express nothing but the relation which holds between the pressures and volumes of a certain quantity of steam, which quantity may be changed by altering the scale of length of the diagrams. Having once realized this, the advantage becomes apparent, in instituting comparisons between a number of engine trials, of taking the common scale of length for the diagrams to be such that they all express the relation between the volume and the pressure of the common unit (1 lb.) adopted for the weight of steam. This common scale is readily obtained by dividing the product of effect-

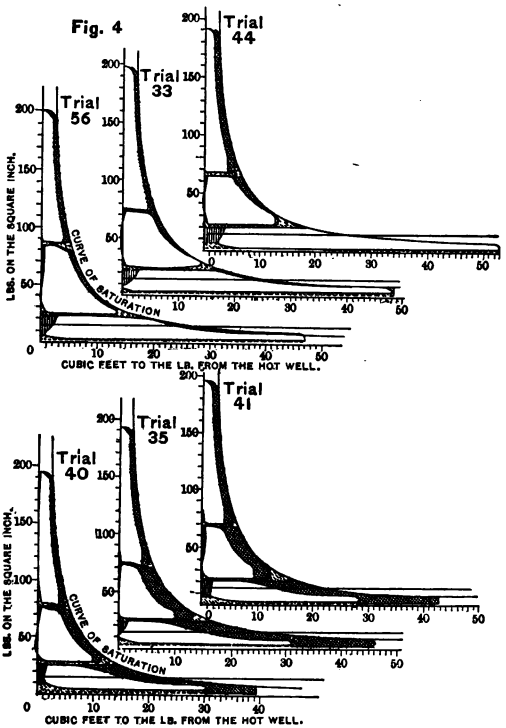
ive volumes, multiplied by revolutions, by the weight of steam passing through the engines per minute, and taking the result as the length of the diagram in any uniform scale; ½ inch to the cubic foot has been that adopted for the first reduction in these trials, the pressures being plotted to 10 lbs to an inch.

The diagrams (Fig. 4) are such mean diagrams, showing the lbs. per square inch pressure and cubic feet volume for each lb. of steam passing through the engines, also Rankine's curve for saturated steam to the same scale. In these diagrams:—

The extreme length of the diagram equals the effective volume swept by the piston for each lb. of steam through the engines.

The distance from the line of zero volume to the expansion or compression-curve at any particular pressure equals the volume of the steam in the cylinder at that pressure, less the steam shut in at compression per lb. of steam through the engine.

Fig. 4


Trial 56
Trial 33
Trial 44
Trial 40
Trial 35
Trial 41
LBS. ON THE SQUARE INCH.
CURVE OF SATURATION
CUBIC FEET TO THE LB. FROM THE HOT WELL.

The area enclosed in the diagram equals the effective work per lb. of steam.

The distance between the compression-line and that of no volume measures on the right equals the volume of initial steam per lb. of steam rendered non-effective by clearance.

The distance between the expansion-line and the saturation-curve equals the volume of steam per lb. of steam through the engines absent on account of condensation, priming and leakage.

The ratio of the horizontal distances from the line of zero volume to the curve at cut-off and release equals the effective ratio of expansion.

The clearness and simplicity of the comparison which these diagrams institute between the areas actually occupied, and those which would have been occupied had the steam been saturated, renders it possible, as well as desirable, to state exactly in what relation the areas stand as regards the theory and economy of the engine.

The area enclosed between the limits

of pressure and volume by the line of zero volume, the line of condenser pressure, and the saturated curve, expresses in foot-lbs. the greatest possible amount of heat that can be converted into work, through the agency of 1 lb. of steam maintained in a state of saturation between these limits. The areas included in the measured diagrams represent the heat which has been so converted by the agency of each lb. through the engines, and the various intervening areas represent loss in conversion.

These are facts which it is important to bear in mind in dealing with jacketed engines, in which 1 lb. of steam through the engines does not represent a certain quantity of heat, which will be the same whether it is realized or not. For such engines it is impossible to make the diagrams represent the comparative efficiencies, actual and theoretical. With unjacketed engines the case is different, as the lb. of steam represents, at a particular pressure, a definite quantity of heat through the engine, however much of it

is converted, and if a special adiabatic line be substituted for the saturated line, the relation of areas will be the relation of efficiencies. In the present case, however, it seemed better to treat all the diagrams in the same way, and to make a separate comparison of the efficiencies with the highest theoretical efficiency between the same limits. With the unjacketed as well as with the jacketed trials, the theoretical efficiency has been calculated as for saturated steam. This comparison for all the trials is given in Table III.

The Condensation, Priming and Leakage of Steam in the Cylinders, as shown in the Diagrams.

There are two quantities which it is almost impossible to separate by the inherent evidence of the diagrams.

The missing quantity, which is here shown by the horizontal breadth of the black band, may equally well arise from the steam having escaped by the piston, or having been temporarily converted into water.

This much, however, is evident from the diagrams, that with steam in the jackets, in whatever manner the steam has vanished in the high-pressure engine, it has all re-appeared before the end of the stroke in the intermediate engine, and though some of it has disappeared at the cut-off in the low-pressure cylinder, it has re-appeared again before the end of the stroke. Hence it seems that there is no escape of steam by the pistons of these two engines.

The question remains, however, as to whether steam has not escaped by the pistons of the high-pressure engine, and through the valves, during expansion, into the cylinders of the intermediate and low-pressure engines.

Certain differences in the diagrams taken from No. II engine, when working with different cut-offs, suggested that the rider valves were held somewhat off the back of the main valve by the spindle, so that they leaked steam until the pressure in the cylinder was sufficiently lower than that in the steam-chest to spring the spin-

dle and force the valves home. It became particularly evident in the fifty-fifth trial, and then the cover was removed and the conclusion verified. This source of error was put right, and the fifty-sixth trial, as compared with the earlier ones, shows what has been the effect of leakage in these, namely, the breadth of the black band towards the tops of the diagram from No. II engine.

When the covers were last put on, in August, 1888, the cylinders and valve-faces were all in equally good condition, and there has been no leak from the jackets, while the engines were standing with full pressure in the jackets. The regulators opening into the intermediate receivers were made tight in August, 1888, and were not again opened till after the forty-sixth trial. There was then occasion to open them, and as the engines were standing preparatory to starting the fifty-sixth trial, it was seen that steam was leaking into No. II receiver, probably about ½ lb. per minute; as the valve was found to be shut, there

was nothing to be done, so the trial was run; and, as was to be expected, the diagrams from No. I engine show what, considering the circumstances, is an unusually large black band.

In the absence of definite evidence of leakage, the Author concludes that the missing quantity shown by the black band is everywhere due to condensation.

It is not the intention in this paper to endeavor to establish a complete theory of cylinder-condensation; though it may be well to state that, before designing the engines, the theory was carefully considered and formulated, leaving only the arbitrary constants to be determined from the experiments. For anything like a complete determination of these constants, the experiments have not sufficiently advanced; but this is not necessary to show that in the case of a series of cylinders, all jacketed up to boiler-pressure, the law of condensation would be precisely that which is shown in the diagrams.

Whenever the bounding surfaces are

colder than the steam adjacent to them condensation occurs. To prevent condensation it is therefore necessary to maintain all parts of the cylinder surfaces and port passage surfaces at a temperature at least as high as that of the initial steam.

To do this, in the case of expansion, it is not sufficient (as seems to be commonly assumed) to keep the outside of the metal constituting the walls and covers, merely at the temperature of the initial steam. That, of course, would be sufficient if there were no condensation other than what results from the temperature of the surfaces.

Forty years ago no such other cause of condensation was known. It was revealed, however, by the discoverers, Rankine and Clausius, in 1849, that the expansion of steam reduces its temperature below that corresponding to saturation unless some of the steam is condensed. The manner of action of this supersaturation, caused by expansion in absorbing heat from the walls of the

cylinder maintained at a higher temperature than the steam, does not appear to have been yet ascertained with any degree of certainty; but it is certain that steam in this state of supersaturation does absorb heat with immense rapidity, when the walls are at a higher temperature than the expanded steam. Also the amount of heat necessary to prevent supersaturation is definitely known, though it is, perhaps, well to recall the fact that it is not, even approximately, the heat equivalent of the work done by the steam during expansion.

If the walls of the cylinders are maintained at the temperature of the initial steam, the expanding steam will absorb heat. This heat must pass through the walls; and as heat only flows through metal down the gradient of temperature, the temperature on the outside must be greater than that on the inside. Hence it follows that either the steam in the jackets must be hotter than the initial temperature of the steam in the cylinder, or the mean temperature of the

internal surface of the cylinder will be below that of the initial steam, in which case there will be cylinder-condensation.

How important this degradation of temperature through the walls is, will, perhaps, best be rendered apparent by stating an actual case.

In expanding 1 lb. of steam from a pressure of 203 lbs. to a pressure of 79.3 lbs., the heat per lb. necessary to prevent supersaturation is

55.1 T. U.,

or about 5 per cent. of the total heat in the initial steaming. In a cylinder passing 600 lbs. of steam per hour, to prevent supersaturation there should pass through the walls of the cylinder

33,060 T. U.

Now the jacketed surface of the cylinder of the H. P. engine is less than 1.5 square foot, and the thickness of the metal is more than 0.4 inch. Hence the heat would have to flow through this thickness of metal at a rate of 22,000 T. U. per square foot per hour. From

the known laws of conductivity of iron, this would require a difference of temperature of 38° Fahrenheit.

Thus it appears that, to prevent supersaturation, the temperature of the steam in the jackets of No. I engine must be 38° higher than the mean temperature of the internal steam; or, in other words, that the mean temperature of the internal surfaces will be 38° lower than that of the initial steam, which is at the same temperature as that in the jackets.

What amount of surface-condensation this difference of temperature would cause may be, to some extent, inferred by comparison with the difference between the mean temperature of the surfaces and that of the initial steam when the jackets are empty. Here the initial temperature is about 383°, and that of the exhaust, 302°; the mean, 342°; difference of mean and initial, 41°; so that in this engine the mean temperature of the walls would only be affected to the extent of about 3° Fahrenheit by the

jackets, supposing the whole of the heat to prevent supersaturation were supplied by the jackets. But this would not be quite the case, as some heat is obtained from the difference in the heat given up and absorbed by the cylinder-condensation; and there is no proof that the steam may not be discharged with a certain degree of supersaturation.

However, the reasoning leads to the conclusion that, with steam at initial pressure in them, the jackets would produce a comparatively small difference on the cylinder-condensation in this engine when passing 10 pounds of steam per minute.

In No. II, the intermediate engine, the case is different. Here the heat which has to flow into the cylinder through the walls is nearly the same as in No. I; but the surfaces are double as large and of the same thickness, so that the fall of temperature would be about one-half, or 20°. The temperature of the steam in the jackets is 81° above that of the initial steam, and the internal walls

would still be 60° above the initial temperature. Hence there should be no condensation on those surfaces which are jacketed. Still there are in this engine, although much less than is usual in jacketed engines, portions of the surfaces which are not, so to speak, jacketed, mainly the surface of the ports and of the piston; and though these derive heat from the jackets, it is through a much greater thickness of metal, and hence would require a much greater difference of temperature to prevent condensation. Thus, even with the jackets at a temperature of 60° above that of the steam, there should probably be some initial condensation.

In No. III engine the jackets have a temperature of 140° above the steam, hence the initial condensation should probably be much less than in No. II.

The diagrams show that this is the case. They exhibit a little condensation, which seems to increase from cut-off until the expansion reaches about 1.5 or 2, and it then diminishes to zero. The

increase after cut-off may be owing to the inertia of the indicator piston depressing the curve, as the springs used have always been as weak as possible, on account of the low pressure.

They also demonstrate conclusively, with such jacketing as there is in these cylinders, that a temperature of 140° in the jackets above the initial temperature is sufficient to prevent sensible cylinder-condensation with as much as 720 pounds of water per hour passing through the cylinders.*

The diagram for the trials 41, 35, 40, show the condensation when the jackets are empty. These three diagrams are from trials as nearly as practicable corresponding in power with those with the jackets on. They are reduced to show the volume per pound of water through each

* In an abstract of the paper the author makes the statement that "In No. II engine, with 80° Fahr. difference in the temperature of the jackets and that of the initial steam, the condensation was reduced from 30 per cent. to 5 per cent., and a difference of temperature of 180° Fahr. between the jackets and the initial steam in engine No. III entirely prevented condensation.—[Editor.]

engine, and the outside curve is the saturation-curve for 1 pound of steam; the horizontal breadth of the black band, therefore, represents volume of steam missing. This includes the volume missing on account of the condensation resulting from expansion in each cylinder as well as on account of cylinder-condensation. It is to be noticed, however, that the steam probably entered each steam-chest dry, so that the only water in excess of cylinder-condensation is that resulting from expansion in that cylinder. This would be represented by a curve drawn from the points in the saturation-curve corresponding in pressure to the points of cut-off, and gradually diverging inwards from the saturation-curve, until at release the horizontal divergence should be about 5 per cent. of the horizontal breadth of the white diagram at that pressure.

The great excess of condensation in the intermediate cylinder over the high-pressure, and in the low-pressure cylinder over the intermediate, is very appa-

rent. This fully explains the difference in the relative speeds of the engines with and without the jackets already mentioned, the speed of No. III compared with No. I being as 1.5 with jackets to 1 without jackets.

The distributions of condensation are very similar in the three cylinders. The ratios which the steam condensed bears to the steam passing through the engines at cut-off, middle stroke, and release are shown in Table IV.

The testing of the boiler was carried only so far as was necessary to check the results of the trials. No chemical tests were taken of the air or coal.

The coal used was Nixon's Navigation mixture, weighed as it came from the heap in the boiler-house. In most of the trials the feed was carefully measured, with the result, already mentioned, that it was from 5 to 10 per cent. greater than the discharge from the hot-well.

Taking the feed, these trials show that the boiler generally evaporated 10.4 lbs. of water per lb. of coal with the press-

ure 195 lbs. and the feed at 130°. This, if all the water were evaporated, would give 11,350 units of heat per lb. of coal.

The temperature at which the gases left the boiler was 500°, and after passing the water heater 250°, the rise of temperature in the water-heater being about 100°.

The source of the loss of water was not discoverable, so that it was not possible to determine whether it escaped as water or steam; and until this point could be determined it was impossible to say from observations on the boiler what the quantity of heat obtained in the boiler might be. The results in Table I are therefore confined to the steam received by the engines.

TABLE I. MEAN RESULTS OF TRIPLE-

	State of the steam-jackets..... {
1.	Number of the trial.....
2.	Date of the trial.....
3.	Time of trial..... {
4.	Lbs. on the square inch mean absolute pressure in the boiler.....
5.	" " " " " " " receiver No. I...
6.	" " " " " " " " No. II..
7.	" " " " " " " " No. III.
8.	" " " " " " " condenser.....
9.	Lbs. on the sq. in. mean effective pressure indicated in engine No. I..
10.	" " " " " " " No. II.
11.	" " " " " " " No. III
12.	Revolutions per minute of engine No. I. .....
13.	" " " No. II.....
14.	" " " No. III.....
15.	I.HP. engine No. I.....
16.	" " No. II.....
17.	" " No. III .....
18.	Lbs. × feet load on brake No. I.....
19.	" " " " No. II.....
20.	" " " " No. III.....
21.	Brake HP. engine No. I with intermediate shaft. .....
22.	" " " No. II " " " .....
23.	" " " No. III " " " .....
24.	Mechanical efficiency of engine No. I with intermediate shaft.....
25.	" " " No. II " " " .....
26.	" " " No. III " " " .....
27.	Total I.HP.....
28.	Total B.HP.....
29.	Mechanical efficiency of engines I, II, III with intermediate shafts..

EXPANSION TRIALS ON SEPARATE BRAKES.

Cylinder jackets at boiler-pressure. Receiver jackets " "			Cylinder jackets empty. Receiver jacket sat boiler-pressure.		
44	33	56	41	35	40
Feb. 12, '89	Dec. 4, '88	April 2, '89	Jan. 29, '89	Dec. 11, '88	Jan. 22, '89
9.52 to 3.51 P.M.	9.45 to 4 P.M.	9.57 to 2 P.M.	9.54 to 3.55 P.M.	9.54 to 4.6 P.M.	10.32 to 2.37 P.M.
200.0	201.0	207.0	205 0	206.0	203 0
199.0	198.0	203.0	204 0	205.0	201.0
65.0	74 0	85.0	67 0	73.0	78.0
21.8	22.7	22.6	21.7	23.1	25.6
1.5	1.7	2.2	1.3	1.8	2.5
73.7	70.53	71.1	72.61	73.8	75.56
29 12	28.99	33.27	30 66	28.2	29.81
12.02	11.71	12.52	12.15	11.19	11.61
115.0	206.0	230.5	146.0	229.0	322.0
135.0	241.0	298 0	127 0	215 0	320.0
152.0	249.0	299 0	109 0	184.0	276.0
8 06	13.82	15.6	10.07	16.11	23.11
9.85	17.54	24.8	9.73	15.17	23.86
15.32	24.4	31.7	11.12	17.23	26.90
300.0	300.0	300 0	300 0	300.0	300.0
320.0	320.0	320.0	320.0	320.0	320.0
400.0	400.0	420.0	400.0	400.0	400.0
6·56	11.74	13 13	8.32	13.05	18.35
8.21	14.65	18 12	7.72	13.07	19.46
11.55	18.93	23.9	8.29	13.98	21.00
0.812	0 848	0.844	0.825	0.81	0.794
0.835	0.835	0.73	0.793	0.861	0.815
0.753	0.775	0.754	0.75	0.81	0 780
33.23	55.76	72.1	30 92	48.51	73.87
26.32	45.32	55.15	24.33	40.1	58.81
0.792	0.813	0.765	0.79	0 826	0 80

TABLE I.—

State of the steam-jackets.....

Number of the trial.....

30. Lbs. per minute of water passing through the condenser.....
31. Degrees Fahrenheit initial temperature " "
32. " " final temperature " "
33. T.U. per minute taken up by " " "
34. T.U. discharged as I.HP. per minute.....
35. T.U. per minute discharged as I.HP. and in condensing water.....

36. Thermal efficiency as given by heat discharged in condensing water.

37. Lbs. water discharged from hot-well per minute. .....
38. Degrees Fahrenheit temperature in the hot-well .....
39. " " " " " above temperature of feed
40. T.U. per minute discharged from the hot-well.....
41. " " " by radiation.....
42. " total per minute discharged from engines.....

43. T.U. necessary to evaporate 1 lb. of feed to economizer.....
44. " per min. received as dry steam (= lbs. discharged from hot-well)
45. " received from each lb. of water condensed at boiler pressure..
46. Lbs. per minute of water from the jackets, &c.....
47. T.U. per minute received from the water from the jackets, &c.....
48. Total T.U. received per minute by the engines.....

49. Degrees Fahrenheit temperature of feed to the economizer.....
50. Degrees added to temperature of feed in economizer.....
51. Lbs. per hour of water discharged from the hot-well.....
52. T.U. " received by water " " " in the economizer

53. Lbs. jacket-water per hour returned to the feed-pipe into the boiler
54. Degrees Fahrenheit temperature of the boiler. .....
55. Degrees mean temperature observed after mixture with feed.....
56. Lbs. per hour of mixed feed to boiler.....
57. T.U. to evaporate 1 lb. of mixed feed.....
58. " per hour taken up in the boiler.....

59. T.U. per hour received from the furnace (exclusive of steam lost)...

Continued.

Cylinder jackets at boiler-pressure. Receiver " " "			Cylinder jackets empty. Receiver " at boiler-pressure.		
44 Feb. 12, '89	33 Dec. 4, '88	56 April 2, '89	41 Jan. 29, '89	35 Dec. 11, '88	40 Jan. 22, '89
209.5	434.0	453.1	264.0	337.5	403.0
49.9	68.0	55.45	51.83	64.0	52.26
79.83	91.03	83.87	83.03	98.03	94.94
6,271.0	9,994.0	12,862.0	8,233.0	11,484.0	17,200.0
1,421.0	2.384.0	3,085.0	1,325.0	2,074 0	3,158.2
7,692.0	12,378.0	15,947.0	9,561.0	13,558 0	20,358.0
0.185	0.192	0.194	0.141	0.153	0.155
5.83	9.58	12.34	7.73	11.47	17.58
102.0	114.0	112.7	108.0	114.0	132.5
22 0	0.0	28.9	25.0	0.0	0.0
128.0	0.0	356.0	200.0	0.0	0.0
1,227.0	1,394 0	1,176.0	503 0	897 0	1.170.0
9,047.0	13,772.0	17,479.0	10,264 0	14,455.0	21,528.0
1,151.0	1,117.0	1,147 0	1,148.0	1,117.0	1,038.0
6,710 0	10,700.0	14,154.0	8,874.0	12,812 0	19,303.0
844.0	844.0	842.0	843 0	843.0	843.0
2.77	3.64	3.95	1.65	1.95	2.64
2,337.0	3,072 0	3,325.0	1,390.0	1,643.0	2,225.0
9,047.0	13,772.0	17,479.0	10,264.0	14,455 0	21,528.0
80.0	114.0	83.8	83.08	114.0	132.5
95.5	83.8	130.7	98 5	76 1	107.7
349.8	574.8	740.4	463 8	688.2	1,054.8
33,500.0	48,200.0	96,800.0	45,700.0	52,500.0	113,600.0
166.2	218.4	237.0	99.0	117 0	158.4
383.0	383.0	383.0	383.0	383.0	383.0
244.0	250.0	257.0	218 0	219.0	260.0
516.2	793.2	977.4	562.8	805.2	1,213.2
987.0	981.0	974.0	1,0 3.0	1,012 0	971.0
509,300.0	778,100.0	951,900.0	570,100.0	814,800.0	1,178,000.0
542,800.0	826,300 0	1,048,700 0	615,800.0	867,300.0	1,291,600.0

TABLE I.—

State of the steam jackets.....

Number of the trial.....

60. T. U. per lb. of coal taken up in the economizer.....
61. " " " " boiler.....
62. " " " total.....

63. T. U. per I.HP. per hour for radiation.....
64. " " " for the engines.....
65. " " " total.....

66. } Lbs. per I.HP. per hour of feed-water to supply as dry steam the heat for... ..... { radiation...
67. { the engine...
70. { total.....

71. } Lbs. of coal per hour (Nixon's Navigation mixture) for. .... { radiation .....
72. { working the engines.
73. { total.....

74. Lbs. of coal per I.HP. per hour for radiation.....
75. " " " " working the engines.....
76. " " " " total.....

TABLE II.

——		Jackets at Boiler-Pressure.			Jackets Empty.		
Number of the trial.....		44	33	56	41	35	40
Thermal unit from the condenser per lb. of water from the hot-well.....	Calculated	1,011	1,014	1,011	1,014	1,009	990
	Measured.	1,075	1,043	1,042	1,065	1,001	978
	Differ'ces.	—64	—29	—31	—51	8	12

Continued.

Cylinder jackets at boiler-pressure. Receiver " " "			Cylinder jackets empty. Receiver jackets at boiler-pressure.		
44 Feb. 12, '89	33 Dec. 4, '88	56 April 2, '89	41 Jan. 29, '89	35 Dec. 11, '88	40 Jan. 22, '89
670.0	646.0	1,006.0	898.0	651.0	950.0
10,186.0	10,436.0	9,895.0	9,961.0	10,103.0	9,850.0
10,856.0	11,082.0	10,901.0	10,759.0	10,754.0	10,800.0
2,216.0	1,500.0	978.0	976.0	1,110.0	950.0
14,120.0	13,320.0	13,567.0	18,969.0	16,768.0	16,535.0
16,336.0	14,820.0	14,545.0	19,945.0	17,878.0	17,485.0
2.0	1.34	0.84	0.84	1.0	0.86
12.2	11.92	11.83	16.45	15.0	15.04
14.2	13.26	12.68	17.3	16.0	15.9
7.0	7.5	6.4	2.8	5.0	6.4
43.0	67.06	89.8	54.44	75.65	113.2
50.0	74.56	96.2	57.24	80.65	119.6
0.21	0.13	0.09	0.09	0.10	0.09
1.29	1.24	1.24	1.76	1.56	1.53
1.50	1.33	1.33	1.85	1.66	1.62

TABLE III.—RELATIVE AREAS of DIAGRAMS per LB. OF STEAM THROUGH the ENGINES, and THERMAL EFFICIENCIES OF ENGINES.

	Number of trial.....	44	33	56	41	35	40
1	Theoretical area, ft & lb.	238,645	233.545	228,420	235,500	233,000	221,860
2	Measured area "	188,056	192,067	192,000	127.545	139.546	144,250
3	Percentage of theoretical area.....	79.0	82.0	84.0	54.0	60.0	65.0
4							
5							
6							
7							
8	Theoretical efficiency, p c	23.3	23.2	22.7	23.3	23.2	22.4
9	Measured efficiency, p. c.	18.5	19.2	19.4	13.8	15.3	15.5
10	Percentage of theoretical efficiency	79.4	82.6	85.4	59.2	65.9	69.4

TABLE IV.—CONDENSATION WITHOUT JACKETS.

—	Number of the Trial.	Revolutions per Minute.	Ratio of Expansion.	Proportion of Total Steam condensed at		
				Cut-off.	Mid-Stroke.	Release
Engine No. I.....	41	146	2.7	0.40	0.39	0.30
	35	229	2.3	0.29	0.27	0.22
	40	322	2.0	0.22	0.21	0.17
Engine No. II.....	41	127	2.4	0.41	0.345	0 29
	35	215	2.4	0 38	0.34	0.26
	40	320	2.2	0.30	0.27	0.14
Engine No. III.....	41	109	2.7	0.51	0.48	0.37
	35	184	3.05	0.48	0.47	0.33
	40	276	2.6	0.32	0.36	0.23

Discussion.

Mr. E. A. Cowper felt that, had he been trying the experiments himself, he should have preferred to make the engines run at one speed, and then the indicator diagrams would have been comparable without any alterations. The diagrams shown, he understood, were average diagrams from a large number of experiments, but altered in proportion to the speeds. Of course, in making such experiments the engine must be in good order; he did not say perfect order, because that was not to be attained, but it must be really good. Unfortunately, up to experiment No. 55 it seemed that the expansion-valve was off the face, viz., that the slide-rod sprung it off the face, and it was not until the steam-pressure was sufficiently low in the cylinder that the valve came up to the face. It was stated in the paper that there were leaks of steam into the receivers; he should

therefore have preferred putting aside those fifty-five experiments and to have gone on to the fifty-sixth, and to have taken a number of others afterwards when the engine was in good order. There seemed to have been an escape varying from 5 to 10 per cent. of feed-water, which should have been stopped before calculating the exact quantity of water that went into the engine. The reverse had been done by the Author, who had taken into account the water coming from the engine, and the calculation of fuel, he understood, was upon that. The fuel for all the steam evaporated from the boiler had to be paid for, and, therefore, engineers were in the habit of estimating the feed water; but as neither the weight of fuel burnt, nor the quantity of feed-water used, was given in this case, it was impossible to follow out the calculation. He must take exception to the statement that the steam was put into the cylinder as dry as possible, because part of the steam supplied to the engine went first through

the jackets. Now the jackets acted to a certain extent in condensing steam, and thereby did very useful duty, keeping the cylinders warm. The steam coming into the engine passed from the boiler dry, but what passed through the jackets became damped. The Author stated that the jackets were made so that the water should drain back to the boiler. The engines were arranged in such manner that the water-line of the boiler was considerably below the water in the separator, the object being that any water that went into the separator should be allowed to return to the boilers by gravity, and there was a head of 6 feet to effect this. But, as Mr. Cowper had pointed out on former occasions, if the jackets were connected with the boiler to take steam from the boiler, and the bottom of the jackets also connected to the boiler again and the water was allowed to run back, all the air that came over with the steam was lodged in the jackets. In this case the damp steam from the jackets would probably bring

most of the air out with it; but the fact that "the jackets were liable to fall off in efficiency" rather pointed to their not having pure high-pressure steam in them. This was not a fanciful objection, but was one practically of great importance, and he always made a point of letting out the air occasionally, or else continuously, from the jackets, so that no air should be lodged in them. Air, of course, would render them non-operative to a certain extent. He approved of the steam-jacketed receivers between each pair of cylinders, having found out the advantage of that arrangement in 1856, when he first introduced them. Cutting off the steam before the half-stroke in each cylinder prevented any second gush of steam into the cylinder whilst the piston was making its stroke, and before the cut-off, and was also very good. The engine, however, not being coupled, interfered somewhat with this, inasmuch as with that arrangement the gush of steam might come from the engine at different times, or just before the mo-

ment of cutting off steam. The indicator diagrams of course must vary a little, because of the varying time of one cylinder taking steam from the other; not to a great extent in this case, since the steam-jacketed receivers were so very large, but it should be avoided, because the steam should not enter without doing its full duty. He could not but think that there must be some rather considerable friction in the engine, or else the duty would be greater with 185 pounds steam and triple expansion. Higher duty had been attained with 55 pounds steam in a compound engine with only two cylinders, but of large size.

Professor W. C. Unwin said that the Author drew an ideal compression curve, and then moved back the diagrams until the compression curve came to the vertical axis of the co-ordinates. It was a matter of small importance. He thought that when the clearance spaces were left in the diagram rather more was shown, and rather fewer assumptions were made, and it lent itself better to

adding to the diagram the rectangles, showing the heat exchange with the cylinder walls. He might add that the diagrams he obtained were almost identical with the high-pressure and the intermediate cylinder diagrams. There was no third cylinder in the Guilds engine, and therefore not quite as great re-evaporation; but when the jacket was in full action, a low-pressure diagram was obtained just touching the saturation curve in the best cases, or slightly off it in the others. He had for some time noticed how very large the jacket condensation was. In trying the Worthington engine at the Hampton Water-Works in conjunction with Mr. Mair-Rumley, from 13 to 16 per cent. of the whole of the steam used was condensed in the jackets. In the Guilds engine the amount of condensation was often 12 per cent., the difference in the two cases being that there was no jacket on the receiver between the cylinders, whereas in the Worthington engine, at Hampton, there was a jacket on the

intermediate receiver. The Author got still larger condensation from the jackets of the Owens College engine. It was rather remarkable that it was worth while throwing away from one-fourth to one-fifth of the steam, in order to keep the cylinder warm. He had looked with interest at the theoretical considerations on the action of the jackets on the cylinder submitted by the Author, who had taken a new line in considering merely what amount of heat supplied from the jackets was necessary to re-evaporate all the water produced by adiabatic expansion. It rather seemed that, in the numerical figures, the Author overlooked the fact that the whole heat must be given to the steam during a half of the revolution. The Author neglected entirely what he believed in many engines was an important thing, namely, the percentage of priming water. He did not think it was ever absolutely absent, and suspected that, in a good many cases, it was an important factor in cylinder condensation. Then the mean

temperatures, and the total quantity of heat transmitted to the steam were dealt with. It must be remembered that, in order to prevent condensation on the cylinder walls, their temperature must be at every point of the stroke higher than the temperature of the steam, and that again would introduce a correction into the numerical result. The extraordinary cumulative effect due to the presence of water on the cylinder walls could not be more strikingly shown than by the very large black patches on the indicator diagrams for the case of no jackets.

Mr. W. W. Beaumont considered the paragraph relating to "trials without jackets" somewhat obscure. He understood, however, that it meant trials without jackets round the cylinders and the cylinder covers; but still with the jackets at work round the steam-pipes or the receivers. The question that occurred to him was this: the Author stated that the steam was admitted into the low-pressure cylinder as dry

saturated steam, carrying into the cylinder the total heat of evaporation from the temperature of the condenser at the temperature of admission, and that it carried this heat to the condenser, less the heat-equivalent of the indicated work done in the cylinder. Was it right in that case to assume that the whole of the heat-equivalent of the work done should be deducted, or only that part of the work-equivalent which represented the expansion part of the stroke? It appeared that the receiver was actually acting as a boiler, and that instead of deducting from the heat carried away into the condenser the whole of the work represented by the whole of the stroke, only that part represented by the expansion part of the stroke should be deducted, because steam was actually evaporated so as to suit the assumption that perfectly dry steam entered the cylinder—evaporated in that vessel just the same as if it were a boiler. He thought, therefore, that the heat-equivalent of the work done during

the admission part of the stroke should not be taken into consideration as a deduction from the heat passing into the condenser; in other words, that the $p\,v$ part of the stroke should be deducted. That would slightly alter the expression given, which would become

$$\frac{H_1 - h_s}{772} - \left(\frac{(\text{I.HP.} \times 42.7) - \frac{pv}{772}}{\text{lbs. per minute from the hot-well}} \right)$$

If that was not deducted he found that the equivalent in heat-units of that part was very nearly the heat-equivalent which the Author said in Table II was in excess, namely, 60 units. It made rather more than the 60 units, and less heat should be accounted for in the observed than in the calculated figures. The next point was, that the Author, in the arrangement shown, used high-pressure steam to evaporate water from which he got low-pressure steam. It was obvious that there must be a loss in doing that. If the paragraph was taken to mean that the trials really were without jackets altogether, without jackets round

the pipe as well as round the cylinder, then, of course, the same correction would be true, inasmuch as it would not be the total heat of evaporation that would have to be taken into consideration, but the heat of evaporation under constant volume only. The Author gave categorically his reasons for adopting the new instead of the older form of brake. Firstly, he said that such brakes required constant observation and watching; secondly, that a single engine could not be started without relieving the load; and thirdly, that brakes were cumbersome and not easily adapted to measure greatly different powers. With reference to these three points it was fairly shown in the paper that Mr. Beaumont had the honor of presenting to the Institution last session, that friction-brakes not only could be, but had been used, for trials of engines extending over greater periods than any of the trials mentioned in connection with this engine; that engines could be started without relieving the load, and that the variation in the position of the

suspended load need not be very great. Taking the last point first, he might recall attention to the brake known as the Balk brake, used for many years by Messrs. Ransome, and for engines working at very different powers. That brake could be set so that it should be run at least an hour at a time with the weight in the scale-pan, not moving more than $\frac{3}{4}$ inch, less the diameter of the pin that went into a slot of that width. With that brake the work upon the engine, or that the engine was doing, could be read off direct at any time, and the brake was run without water. Another and a better brake for varying loads, and needing perhaps even less attention, was one used by Mr. Halpin and others, in which there was a channeled fly-wheel with water running into it. The water simply ran into it and evaporated. If the wheel was kept sufficiently cool, that brake might run for almost any length of time without more attention than the one man driving and stoking the engine could give it. The variation in the level of the

load was very little indeed. With regard to that variation he noticed that the Author had shown in the illustration something very much like a dash-pot, and above that again, a spring, and he said the danger of such an arrangement had been carefully considered. This was with reference to the water-brake. He did not say what the precautions were, but he supposed they took the form of a dash-pot. If a dash-pot were used with either the Balk brake or the water-channeled wheel-brake, or the rope-brake on the water-channeled wheel, all the results could be as easily, as economically and as accurately obtained as with this water-brake, and with a smaller number of pipes, water-connections and so on.

Mr. G. R. Bodmer said, with respect to the trials themselves and their results, it was rather to be regretted that the engines were allowed to run at different speeds, because it prevented the conclusions drawn from the experiments being directly applied to ordinary triple-expansion engines. For instance, when the

low-pressure ran at a higher speed than the high-pressure engine, it was equiva-

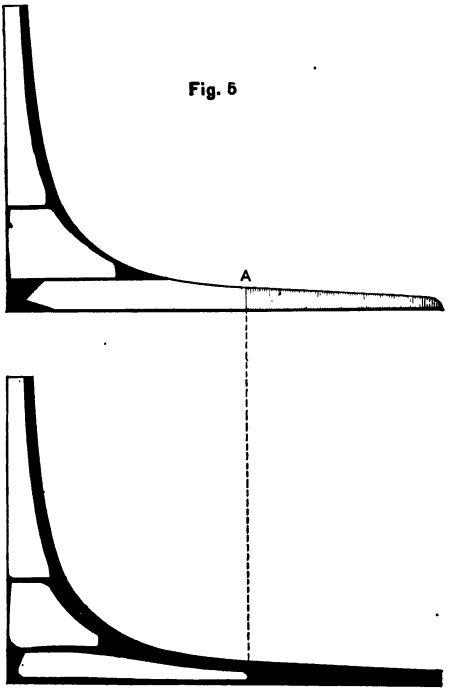


Fig. 5

lent to having a larger low-pressure cylinder, viz., to having a greater amount of

expansion than was indicated by the cylinder volume. It was obvious that in an ordinary engine it was impossible to vary the final cylinder-volume, and, with a given cut-off, the expansion ratio was fixed. On that account it would be wrong to infer that the same advantage could be secured by jacketing an ordinary triple-expansion engine or compound engine as the Author had been able to obtain by jacketing his engine. To explain his remarks he had prepared *Fig.* 5, from the Author's illustrations. The lower diagram represented the experiments with the cylinders unjacketed; the upper diagram represented the corresponding experiments with cylinders jacketed. Taking the lower figure to represent the diagram of an ordinary unjacketed triple-expansion engine (the length of the diagram corresponding to the stroke of the low-pressure cylinder), then, if that steam-engine were jacketed, the volume of the low-pressure cylinder could not be increased or treated in a way that was equivalent to increasing it,

by allowing it to run at a greater speed than the high-pressure part of the engine; therefore, the advantage obtained from jacketing that engine would be represented, not by the whole of the upper diagram, but only by the white portion; and, as compared with the Author's results, the whole advantage represented by the shaded portion of the diagram was lost, being the equivalent of the increased speed of the low-pressure, as compared with the high-pressure engine. With this portion of the diagram cut off it was evident the advantage obtained by jacketing was much diminished. Nevertheless, the Author's experiments were extremely interesting, and tended to show that, in order to obtain the full advantage from jacketing an engine, the volume of the low-pressure cylinder must be made greater than if the engine were unjacketed, assuming a given cut-off in the high-pressure cylinder. Coming to the question of the action of the jackets, the Author assumed that heat actually passed through the walls of the cylinder

from the steam in the jacket to the steam expanding in the cylinder, and attributed, at any rate, part of the beneficial action of the jacket to the passage of such heat. The figures appeared to a great extent to contradict this view, and Mr. Bodmer had prepared a "Supplementary Table," recapitulating a few of the Author's figures, and also giving some ratios calculated from them. Before referring to that Table he might mention that recent experiments had shown, notably those of Mr. Willans, and also the Author's trials with the unjacketed engine recorded in Table IV, that it was not essential to re-evaporation that the cylinder should be jacketed; but that re-evaporation took place in an unjacketed cylinder to a certain extent. The necessary heat, of course, came from that previously absorbed by the metal during the admission period. Line III of the Supplementary Table gave the ratio of heat received from the jackets to that received as dry steam, and he had placed the results of cor-

responding experiments with engines jacketed and unjacketed side by side. For instance, column 44 referred to a

SUPPLEMENTARY TABLE.

		44	41	33	35	56	40
I [44]	T. U. received as dry steam per minute	6,710	8,874	10,700	12,812	14,154	19,303
II [47]	T. U. received from the jackets, &c., per minute	2,337	1,390	3,072	1,643	3,325	2,225
III	Ratio of heat received from jackets to heat received as dry steam, per cent	34 8	15.7	28.7	12.8	23.5	11.5
IV	Excess of heat received by jacketed as compared with unjacketed engine, per ct.	16.5	..	14.0	..	10 8	
V [36]	Thermal efficiency	0.1-5	0 141	0.192	0.153	0 194	0.155
*VI	Relative increase of efficiency due to cylinder jacket, per cent	31.2	..	25.5	..	25.1	
VII	Ratio of heat lost by radiation to heat received as dry steam, per cent	18.3	..	13 0	..	8.3	

* These figures, calculated from the data given, are so high as compared with those obtained by other investigators that the results should be received with

series of trials with jacketed cylinders; 41 to the corresponding series with unjacketed cylinders. Taking the first two columns, for the jacketed engine the percentage was 34.8, and for the unjacketed

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some caution, as known leakages and unexplained losses interfere with the value of the results.

One of the most exhaustive experiments to determine the influence of the steam jacket on cylinder condensation was made by Prof. J. E. Denton, and the results were presented to the American Society of Mechanical Engineers in November, 1889 (Vol. XI). The test was made on the Corliss Pumping Engine at Pawtucket, R. I., and covered a period of 72 hours, under the usual working conditions. The average pressure per gauge was with jacket 127½ lbs. and without jacket 124½ lbs. The conclusion reached by Prof. Denton in this case was that "the most that can be claimed for the jacket is that it probably caused no loss, and may possibly have caused a saving not exceeding 3% of the total steam consumption."

Prof. Denton stated in the discussion of his paper that Mr. Geo. Corliss once informed him that he considered 5 per cent. of gain all which could be relied upon from the influence of jackets, and at another time Mr. William Corliss assured Prof. Denton it was difficult to recall instances where there was proof of even this small amount of gain being attributable to jacketing.

Other investigators have placed the saving at between 11% and 12%, but the conditions of the tests when using the jackets were so different from those in which no jackets were used that these results are open to some doubt. [Editor.]

15.7. The 15.7 represented heat received only from the receiver jackets, and, assuming the ratio of the heat lost in the receiver jacket to be the same in both sets of trials, the proportion of heat given off by the cylinder jackets would be represented by the difference between 34.8 and 15.7. Line IV gave in percentage the excess of heat received by the jacketed as compared with unjacketed engines. That was for the first two series of trials (44 and 41), 16.5, for the second (33 and 35), 14.0, and for the third (56 and 40), 10.8 per cent. This was the heat actually given off in one way or another by the cylinder jackets as distinct from the receiver jackets, and owing, presumably, to the expenditure of that heat, the increased efficiency represented by line VI was due. From this it appeared that for an expenditure of heat equivalent to 16.5 an increase of efficiency equal to 31.2 per cent. was obtained. That of itself was sufficient to show that only a portion of the apparent advantage secured came from the heat

passing through the walls. To arrive at the proportion of heat actually transferred from the steam in the cylinder jackets to the working steam, the heat lost by radiation must be deducted from the total expenditure in the cylinder jackets. The Author had not separated the loss by radiation in the receiver jackets from that in the cylinder jackets, and consequently Mr. Bodmer was unable to arrive with accuracy at the latter quantity. So far as could be judged from the illustrations accompanying the paper, the ratio of the surface from which radiation took place for the cylinder jackets to the total radiating surface was as 1 to 3, and assuming this to be correct, an estimate could easily be formed of the quantity passing through the walls, which would be less by from 25 to 30 per cent. than the values given in line IV. As a matter of fact it might not be superfluous to point out that, on thermo-dynamic principles, it was impossible any improvement in efficiency could result from an increase

in the work done, due directly to the transfer of heat from the steam in the cylinder jackets to the steam working in the cylinder, because heat communicated in such a manner was utilized even less effectively than the heat introduced with the steam admitted to the cylinder. From this it was obvious that the whole of the advantage gained by the use of the cylinder jackets was a consequence of the restoration of heat absorbed previous to cut-off by the cylinder walls; this restoration took place partially in an unjacketed engine, but was rendered more complete by the action of the jacket in preventing recondensation during the admission period in the intermediate and low-pressure cylinders. The best of all would be to prevent initial condensation in the high-pressure cylinder, where the steam was of most value. It was to be regretted that the Author had not stated the proportion of steam condensed at various points of the stroke in the trials with jackets, in the same way as this had been done in Table IV for the trials

without jackets. The Author said: "If the walls of the cylinders are maintained at the temperature of the initial steam, the expanding steam will absorb heat. This heat must pass through the walls; and as heat only flows through metal down the gradient of temperature, the temperature on the outside must be greater than that on the inside." Now this statement, he thought, was at variance with known facts. The heat absorbed by the expanding steam did not necessarily pass through the walls. In a cylinder without a jacket, the temperature of the walls after cut-off was lower than the temperature of the initial steam, and yet experiments had clearly shown that the expanding steam absorbed heat, and re-evaporation took place. It was in the main not a question of the passage of heat through the walls, but, as already indicated, of a restoration of heat to the steam from which it had been taken during the admission period. The Author's calculation, of the amount of heat per lb. of steam necessary to prevent

supersaturation, was not applicable to the conditions obtaining in his engine ; it would only be correct if the whole of the steam, at the moment when expansion commenced, were quite dry and were maintained in this condition throughout. The quantity of heat required to prevent the condensation otherwise accompanying work performed (in a non-conducting cylinder) depended on the proportion of water initially present. If about 50 per cent. of water were present, no heat would be required to prevent further condensation, the saturation curve for the dry portion of the working mixture (steam and water) corresponding with the adiabatic curve for the whole of the mixture. The only correct method was to calculate, from the total measured work done during expansion on one side of the piston and the increase (or decrease) in the internal heat, the quantity of heat communicated during expansion, and compare this with the heat given up to the walls during admission. The difference between these

two quantities would give the heat which must pass through the walls from the jacket-steam.

Mr. P. W. Willans said the subject was one which had long been of interest to him.

He had read the paper carefully, checking it figure by figure, so far as the data given would allow, because, first of all, he wished to understand the methods on which the Author had been working. He regretted to say that he had not been favorably impressed with many of the Author's methods, and he also totally disagreed with his choice of the standard with which to compare the work of his engine. He proposed, therefore, to follow the Author through his various methods and calculations, and the data which he would make use of were the following:

Table I, line	56	.	lbs. per hour of mixed feed to boiler, neglecting leakage.
" "	37	.	lbs. of water discharged from hot-well per minute
" "	46	.	lbs. per minute of water from the jackets, &c.
" "	27	.	Total I.HP.
" "	5	.	Initial temperature calculated from pressure in 1st receiver.
" "	6	.	Final temperature calculated from condenser-pressure.

Now with respect to these data, and the methods by which they had been obtained, he wished to offer a few criticisms. The mixed feed per hour (line 56) was the sum of water discharged from the hot-well, and of that discharged from the jackets. Neither of these quantities had been weighed as a whole, and the only reason he could see for the method adopted—namely, weighing 100 pounds at a time in the case of the hot-well water, and noting the rate at which the water rose in the separator for a portion of the time in the case of the jacket-water—was that an engine trial and boiler trial were made together, and that the Author wished to get the best possible results in both. The Author had stated that there was a difference in most cases of from 5 per cent. to 10 per cent. between the water supplied to the boiler and that discharged from the hot-well. He did not think that any experimenter was justified, when he knew of such a discrepancy, and knew also that the boiler was "practically tight," in taking

the discharge from the hot-well for the purpose of calculating his results; he could not believe that there was any law of nature by which differences of 10 per cent. between the two could be accounted for. The Author took no account of what was usually present, namely, a little vapor from the hot-well, and the same from the piston-rods and glands (there were nine rods in all, including the valve-rods); but he jumped to the conclusion that the engine had nothing to do with this 5 per cent. or 10 per cent. Mr. Willans had given, in all cases, in his paper, the total feed-water pumped into the boiler during the period of the experiment, the water being drawn from a large tank mounted on a weighing-machine (entailing one reading only for the whole time), and he had based all his figures upon the total so ascertained. Would the Author think it fair of him if he deducted 5 to 10 per cent. from the total, because the Professor found that such a difference generally existed? He also thought that the figures for feed-water

for each trial might have been supplied, seeing that there was a doubt in the matter. Next, with respect to the figures for jacket-water, Table I, line 46. The Author explained that these were arrived at by "measuring the rate of discharge from the jackets every half-hour." The cock at the bottom of the separator was, as he understood it, shut by the student every half-hour, and the rate at which the water rose in the gauge-glass on the separator was noted. Now the dimensions of the separator were such that not more than 20 or 25 pounds of water could be allowed to collect in it, so that, assuming absolute accuracy in the observations taken every half-hour, about 40 or 50 pounds of water per hour would be measured; while 237 pounds in the case of trial 56 was assumed to be the quantity passing through per hour; if, therefore, he understood the Author correctly, only one-fifth of the jacket-water had in this trial gone through even a form of measurement. Surely it would have been a more certain plan to have

made the engine or boiler trials independently, and to have weighed accurately the jacket-water for the entire trials. Next, as to the total indicated HP., an elementary difficulty was that the dimensions necessary for recalculating the indicated power were not given. A dimension was also missing, for, in calculating the power, however interesting the diameter of crank-shaft might be, it was not so necessary as the diameter of the piston-rod, in getting at the net area of the cylinders, and he trusted it had not been omitted from the calculations as well as from the table of dimensions. Then, if he understood the Author correctly, the mean pressure had been calculated by measuring the pressure on the diagrams at ten breadths, and taking the mean of these as the mean pressure. "These results have been several times checked by a planimeter without establishing any sensible difference." He wished to point out that the measurement could only be accurately done by means of a planimeter, for if measure-

ments by ordinates were taken in the middle of ten equal breadths, in the case of the jacketed low-pressure diagrams, there would be, for trial 33, a result at least 5 per cent. in excess of that given by the planimeter; this was on account of the late admission and the position which the first measurement happened to take with reference to the corners missing from the diagram.

In order to ascertain the thermodynamic efficiency of the engine it was necessary to ascertain the higher and the lower limits between which the engine worked. Here again he was met by a difficulty, for the temperatures given in Table I, line 54, as the temperatures of the boiler, were the same throughout all the six trials, namely, 383° Fahrenheit, whereas the mean absolute pressure in the boiler varied from 200 pounds in trial 44 to 207 pounds in trial 56. This would not make much difference to the efficiency, but it had made it a little difficult to follow the Author's calculations, the temperature being the starting-point.

In going through the paper casually, his attention had been drawn to the very high thermal efficiency which the Author gave for one of the trials, 19.4 per cent. in one case. The percentage of theoretical efficiency actually obtained in this case (trial 56) was given as 85.4. In non-condensing engine trials he had been accustomed to get as much as 82 per cent. himself; but in such condensing engine trials as he had made it was very much below that, and he had not hitherto seen, in any recorded experiments, figures at all approaching 85 per cent.; he had, therefore, from the Author's data named above, recalculated all his figures as to efficiency in the standard methods, and the results of these calculations were given in the annexed Table:

— Number of trial.... .....

A. Lbs. of mixed feed per hour (neglecting leak)—Table I, line 56.....

B. Total indicated HP.—Table I, line 27..... .....

C. Water per indicated HP. per hour, neglecting leak, but otherwise as usually reckoned, *i.e.*, hot-well and jackets.....

D. Water per indicated HP.—Table I, line 70 .....

E. Measured efficiency—Table III, line 9..... .....

F. Thermal efficiency as usually reckoned—Table I, $\frac{\text{line 34}}{\text{line 48}}$..... . .

G. Initial and final temperatures, Fahrenheit—$\frac{T^1 \text{ steam-chest}}{T_2 \text{ condenser}}$.....

H. Cost of 1 lb. of steam in thermal units, water raised from condenser temperature to steam-chest temperature and evaporated at steam-chest temperature..... .....

I. Lbs. per minute feed from hot-well—Table I, line 37..... .....

J. Thermal units received per minute by engine in weight of steam answering to hot-well discharge..... ..... .

K. Thermal units given up per lb. of steam condensed in jackets at boiler temperature.....

L. Lbs. of jacket-water per minute—Table I, line 46..... .....

M. Thermal units received by engine per minute from jacket-steam....

N. Total heat per minute (feed and jackets). ....

O. Heat utilized—Table I, line 34..... .....

P. Absolute efficiency—$\frac{\text{line O}}{\text{line N}}$ .....

44	33	56	41	35	40
516.2	793.2	977.4	562.8	805 2	1,213.2
33.23	55.76	72.1	30.92	48 51	73.87
15.53	14.23	13.56	18.20	13.60	16.42
14.1	13 2	12 68	17.3	16.0	15.9
18.5	19.2	19.4	13.8	15.3	15.5
15.71	17.31	17.65	12 91	14.35	14.67
381.15	380 7	382 8	383.2	383 6	382.0
115.9	120.35	129.8	110.8	122.4	134.6
1,108.9	1,101.3	1,095 5	1,114.6	1,103.0	1,090.5
5.83	9 58	12.34	7.73	11.47	17.58
6,464.8	10,579.2	13,518.5	8,615.8	12,651.4	19,171
843.43	843.1	841.29	841.89	841 59	842.5
2.77	3.64	3 95	1.65	1.95	2.64
2,336	3,068	3,323	1,389	1,641	2,224
8,801	13,648	16,840	10,005	14,292	21,395
1,421.0	2,384.0	3,085.0	1,323.0	2,074.0	3,158.0
16.14	17.46	18.31	13.22	14.51	14.76

—	Number of trial.....
Q.	Theoretical efficiency of Carnot cycle: heat utilized per lb. of steam = latent heat at $A \times \frac{A-B}{A}$ .....
R.	Theoretical efficiency of Clausius perfect engine: heat utilized per lb. of steam $=(1{,}438-0.7A)\frac{A-B}{A}+(A-B)-B\log_{\epsilon}\frac{A}{B}$.....
S.	Theoretical efficiency of jacketed engine: heat utilized per lb. of steam $=1{,}438\log_{\epsilon}\frac{A}{B}-0.7\,(A-B)$.....
T.	Author's theoretical efficiency—Table III, line 8.....
U.	Percentage of theoretical efficiency of Carnot cycle— $\frac{\text{line P}}{\text{line Q}}$ .....
V.	Percentage of theoretical efficiency of Clausius perfect steam-engine — $\frac{\text{line P}}{\text{line R}}$ .....
W.	Percentage of theoretical efficiency of Rankine's jacketed engine — $\frac{\text{line P}}{\text{line S}}$.....
X.	Author's percentage of theoretical efficiency—Table III, line 10 ..
Y.	Feed-water per indicated HP. hour required by Clausius perfect engine.....
Z.	Feed-water per indicated HP. hour which would be raised from T_2 to T_1 and evaporated at T_1 by total heat per minute, line N $= \frac{\text{line N} \times 60}{\text{line H} \times \text{line B}}$.....
..	Efficiency $\frac{\text{line Y}}{\text{line Z}}$ .....

In the above Table T_1 and T_2 are temperature Fahrenheit scale,

44	33	56	41	35	40
31.49	30 93	29.98	32.26	30.92	29.34
28.33	27.86	27.06	28.96	27 8	26.52
25.8	25.4	24.7	26.3	25.4	24.3
23.3	23.2	22.7	23.3	23.2	22.4
51.2	56.4	61.0	40.9	46.9	50.3
56.9	62.67	67.6	45.6	52.1	55.66
62.5	68.7	74.1	50.2	57.1	60.7
79.4	82.6	85.4	59.2	65.9	69.4
8.17	8.33	8.65	7.94	8.35	8.86
14.33	13.29	12.79	17.41	16.02	15.93
56 9	62.67	67 6	45 6	52.1	55.66

and A and B the equivalent temperature Fahrenheit absolute.

In this Table, lines A, B, D, E, I, L, O, T, and X were simply transferred from the Author's Table for comparison. The figures in line C were obtained by dividing those for mixed feed in line A by those for total indicated HP. in line B. The figures in line A were those which took no account of the leak to which reference had already been made, and the calculations based on these figures, therefore, gave the Author the benefit of the doubt, as he did himself. The figures in line C were directly comparable with many of the figures given for water per indicated HP. in other cases of jacketed engines, in Mr. Mair-Rumley's among others. It was not everybody who had the advantage of being able to drain the water back into the boilers in the admirable manner adopted by the Author, and frequently the jacket-water was allowed to drain into the hot-well, and was pumped back into the boiler as part of the feed. The Author's figures, "lbs. per indicated HP. of feed-water to supply as dry

steam the total heat for engines," were given in line D for comparison, and it should be understood that these figures did not represent the total steam passing from the boiler in the shape of steam and returning as water, but such an amount of steam as would be evaporated by the heat abstracted from the total steam passing through the engine and through the jackets. Undoubtedly the Professor was justified, in his case, in looking at the matter in this manner; but it was evident that certain conditions were necessary as to the relative levels of engines and boiler in order to permit the jacket-water so to drain back. In line E were stated the Author's figures for thermal efficiency, the only figures given in the paper for the absolute efficiency of the engine. He was surprised to find that it did not include radiation; and no doubt many of the members had not realized that radiation was excluded. The heat-units lost by radiation in some cases amounted to a very large proportion of the heat equiva-

lent of the indicated power. In the case of trial 44, for instance, the loss by radiation was given as 1,227 thermal units, while the indicated power was equivalent to 1,421 thermal units. If radiation was excluded from such trials, the work done might almost as well be neglected. He had recalculated that line by dividing the figures in line 34, which gave the thermal units discharged as indicated HP., by the figures in line 48, which gave the total heat supplied to the engine. The results were supplied in line F, and were those which were usually taken as representing the absolute efficiency of a heat-engine, namely, $\frac{\text{heat utilized}}{\text{heat supplied}}$. The Author was trying jacketed against unjacketed engines, and it was obvious that the radiation in the former must be greater than in the latter. In excluding the radiation, he missed out one of the most important factors in the comparison between the two engines; the effect of the omission being greater in the case of the jacketed than in the

unjacketed engine, it gave the jacketed engine an unfair advantage. Mr. Willans was not in any way concerned to defend unjacketed against jacketed engines; he only said that in this case it was not a fair trial.

What the Author called thermal efficiency was not, therefore, what engineers were in the habit of calling absolute efficiency, and he thought he should show also that the standard with which the Author had compared his results was not the one upon which engineers ought to look as representing the highest theoretical efficiency. He had recalculated the efficiency of the engine in the various trials, taking as the initial temperature that answering to the pressure in what the Author called the first receiver, the steam-chest of the high-pressure engine, and as the final temperature, that answering to the condenser back-pressure. These temperatures were, in the case of trial 56, 382°.8 and 129°.8 Fahrenheit. The feed-water from the hot-well might be assumed to be raised

in the boiler from 129°.8 to 382°.8, and to be evaporated at the latter temperature; if it happened to leave the hot-well at a lower temperature than 129°.8 Fahrenheit (the temperature answering to the condenser back pressure), that was a circumstance which ought not to tell against the engine as an engine, for the feed-water could, theoretically at least, be raised by the exhaust-steam to the temperature of the latter, but that was the highest temperature to which the feed-water could be raised by waste heat from the engine itself. He therefore assumed that each lb. of water leaving the hot-well represented an expenditure of heat equivalent to that required to raise it from the temperature answering to the condenser back-pressure, to that answering to the pressure in the first receiver, plus the latent heat of evaporation at the latter temperature. In the case of trial 56, the expenditure of heat per lb. of hot-well discharge (line H) was 1,095.5 thermal units, and the lbs. discharged per minute (line L) were

12.34; thus the thermal units debited to the engine per minute on account of steam answering to the hot-well discharge were 13,518.5 thermal units (line J). In the case of the jacket-water only the latent heat was given up, and he therefore debited the engine with the latent heat of each lb. of steam condensed at boiler temperature; in the case of trial 50, the jacket-water (line L) amounted to 3.95 lbs. per minute, and the latent heat being 841.29 thermal units per lb., the heat debited to the engine on account of jacket steam was 3,323 thermal units (line M). The total heat debited to the engine per minute, lines J and M, amounted to 16,841 thermal units (line N) = heat supplied. The heat utilized was the heat equivalent of the work done = 3,085 thermal units (line O). Line P gave the absolute efficiency of the engine, as calculated from the above figures, and it was, for trial 56, 18.31 per cent. All these calculations were based on the quantity of water proved to have passed through

the engine, and did not include the leakage already mentioned.

The next question was as to the engine which should be taken as the standard of perfection. It had been a common thing to take the Carnot cycle, working between the same limits of temperature, as the standard; in this cycle all the heat was assumed to be received at the higher temperature, and the heat necessary to raise the condensed steam, from the lower to the higher temperature, was assumed to be supplied by the conversion of a part of the power generated into heat, by the process of compressing a portion of steam remaining in the cylinder. Not long ago he went very fully into this question,[1] and gave his reasons for not using the Carnot cycle as a standard. These reasons were mainly theoretical, but he thought there was one very strong practical reason why the Carnot cycle would never be carried out in practice, and this was that it would always be cheaper to heat the feed-water

[1] Minutes of Proceedings Inst. C. E., vol. xcvi. p. 239.

by the waste gases from the furnace. In order to heat the feed-water dynamically, some part, however small, of the power of the engine must be used, but as the Author had shown in this case, the furnace gases were always at hand for the purpose, and as their heat would otherwise be wasted, it was better to use it than any part of the power of the engine, however small that part might be. If, however, it was fitting in any case to compare the absolute efficiency of an engine with the absolute efficiency of the Carnot cycle, it would be so in this case, where in some trials almost 33 per cent. of the steam passing through the engines and jackets passed through the latter. All the heat supplied to this jacket-steam was supplied at the higher temperature, and consequently the percentage of work due from it was greater than that due from the heat supplied to the hot-well water. Unfortunately, however, although the heat was supplied in the boiler at the higher temperature to the hot jacket-water, it was transmitted

through the walls of the cylinder to the working fluid under very different conditions. Still, he thought it would be interesting to give in line Q the absolute efficiency of the Carnot cycle working between the same limits of temperature (line G). In the case of trial 56 this efficiency was 29.9 per cent. as compared with 18.31, the absolute efficiency of the actual engine. Line U gave the percentage of the theoretical efficiency of the Carnot cycle reached by the actual engine in trial 56, which was $\frac{18.31}{29.9}$, or 61.3 per cent. of the theoretical possibilities, taking that cycle as the standard.

Now putting on one side the Carnot cycle as an unattainable ideal, and for the practical reasons already given an undesirable ideal, he would next compare the efficiency of the actual engine, in trial 56, with the efficiency of the perfect steam engine of Clausius; that was to say, he would compare the return from heat expended with the return from one pound of steam supplied to an engine in

which there were no losses from passages, leaks, or cylinder-walls; in other words, an engine which took one pound of steam into the cylinder at the higher temperature, expanded it adiabatically to the lower temperature, and expelled it at the lower temperature. It might be said that this was also an unattainable ideal, and, in the sense that no engine would ever be made which absolutely came up to it, it was so. Still it represented the highest aim of the engineer, and it was being steadily approached. In certain classes of engines, certain classes of loss had been practically eliminated; for instance, in low-speed engines there was no appreciable loss from passages, and he himself had shown that, in a high-speed engine, it was possible to approach very closely indeed to the adiabatic curve. Under certain conditions this could be approached, for instance, within 5 per cent.; therefore he always took as his standard the result from the practical but perfect engine of Clausius, and compared his results with it. Line R

gave the theoretical return from such an engine always working between the limits of temperature given in line G. This return was, in the case of trial 56, 27.06 per cent. of the heat supplied, and comparing this return with that from the actual engine, which had been already shown to be 18.31 per cent. of the heat supplied, he found that the relative efficiency of the Author's engine, as compared with the perfect steam-engine of Clausius, was 67.6 per cent. The Author, however, took as his standard an engine which gave a return of only 22.7 per cent. of the heat supplied to it, in the case of trial 56. He compared what he called "thermal efficiency" (which was arrived at by neglecting radiation, one of the most important factors in the case), with something that did not, to Mr. Willans, represent the "thermal" efficiency of any engine or cycle whatever.

It would be easier to follow his argument by reference to *Fig.* 6, which showed the work possible from one pound of steam treated in different ways.

T_1 and T_2 were the temperatures between which the engine worked. The inner curve of expansion was the adiabatic expansion-curve for steam expanding from T_1 to T_2; not "a special adiabatic curve," as the Author said, but the adiabatic curve, for there was only one, if dry steam was supplied. The outer

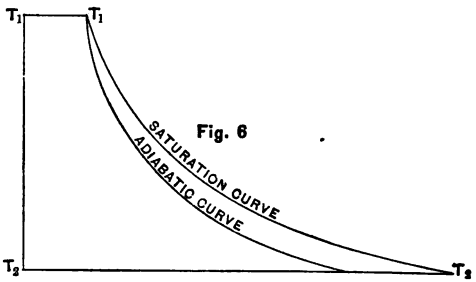

Fig. 6

curve was the saturation-curve for one pound of steam between T_1 and T_2. The inner line was the line which the indicator would trace if no heat was given off to the sides of the cylinder and none received from them. The outer line was what the indicator would trace, if, by adding heat during expan-

sion, the water necessarily formed was re-evaporated by heat received from the steam-jacket. The cost of the inner, or adiabatic diagram for a theoretically perfect steam-engine, was the cost of raising one pound of water from T_2 to T_1, and converting it into steam at T_1. In the case of an engine working between the same limits of temperature as those in trial 56, the cost was 1,055.5 thermal units (see line H). The area of the diagram, namely, the work done, was 297.4 thermal units, representing an efficiency of 27.06 per cent. as given in line R. The cost of the outer diagram was the cost of one pound of steam as above, plus the cost of keeping it during expansion in the state of saturation. This latter added heat was given by the formula $1{,}438 \log\varepsilon \frac{A}{B} - (A - B)$, and was in the case of an engine working between the same limits of temperature as those in trial 56, 259.58 thermal units. The area of the diagram was represented by 335.5 thermal units, so that the return for

heat expended was only 24.7 per cent. Thus, between the thermal limits in question the return from steam held in an engine expanding its steam adiabatically was 27.06 per cent. as compared with 24.7 per cent. in the case of the best possible jacketed engine. The cost of the saturation diagram was to the adiabatic as 1.23 to 1, while the return from it was only as 1.13 to 1; so that when the Author stated that, both in the case of the jacketed and unjacketed trials, he had compared the actual efficiencies with the highest theoretical efficiency between the same limits, and that he had calculated the theoretical efficiency as for saturated steam, it seemed to Mr. Willans to be a contradiction in terms. The work shown by the saturation diagram, compared with the cost of such a diagram, by no means represented the highest theoretical efficiency. The fact was that jacketing was simply a compromise. It was effective in promoting economy in certain cases, because more was gained by preventing conden-

sation than was lost by re-boiling the water formed during expansion, and using high-pressure to make low-pressure steam. But for comparison with the work performed, the standard should be the adiabatic diagram. The question was, how far did it come short of the ideal perfection possible in a steam-engine? Looking at the matter broadly, it seemed to Mr. Willans that the Author had increased the efficiencies in the various trials by omitting factors always of great importance, and in this case of especial importance, and that he had then compared these inflated results with a new standard for which there was no sound reason. The efficiency which the Author gave as 85.4 per cent. was, he considered, 67.6 per cent. These were the figures to which he wished to draw the attention of the meeting. It would be seen that the Author's figures for water per indicated HP., line D, agreed very closely with line Z calculated by him; and he presumed that these slight differences could be accounted for by differ-

ences in the temperatures taken as the higher and lower temperatures for ascertaining the cost of steam. He had given, in line S, the theoretical efficiency of the best jacketed engine working between the limits of temperature in the various trials; but even these efficiencies did not exactly correspond with the efficiencies which the Author had taken as a standard, which were given in line T. Evidently the area of the saturation diagram had not been taken in its fullness as the measure of work obtainable through the agency of one lb. of steam, and the explanation was apparently found in a sentence which appeared in this connection: "The area enclosed between the limits of pressure and volume . . . expresses in foot-lbs. the greatest possible amount of heat that can be converted into work, through the agency of one lb. of steam maintained in a state of saturation between these limits." In fact, the Author had compared his results, not with the whole area of the diagram for one lb. of steam maintained in a state of

saturation between the limits T_1 and T_2, but with that diagram shortened so as to correspond with the terminal pressure in his own experiments, that was, with a still lower standard than the jacketed engine; this was an interesting comparison for some purposes; but had no connection with what was now usually understood by the words thermal efficiency.

With respect to the general results of the trials, a point which had struck him very much was the small effect of the jackets on the missing quantity in the high-pressure cylinder. He should be glad to know whether the Author could give the figures for the condensation in the various cylinders in the case of the jacketed-engine trials, in the same way as he had given them for the unjacketed ones (Table IV). So far as could be judged from the diagrams, the effect of jacketing on the amount of the missing quantity in the high-pressure cylinder was very small. Was it possible that there was considerable leakage past the low-pressure and intermediate jackets? The

mean temperature of the liners of the cylinders would probably be, in these cases, considerably below the temperature of the main casting of the cylinder when the engine was at work. Consequently the liners would be relatively less in diameter and shorter than when the engine was standing, and such leakage would be exceedingly difficult to detect, because the only way apparently of observing it would be when the covers were off and the engine standing, at which time the liners would be as hot as the cylinder. Mr. Bryan Donkin's recent experiments[1] showed that the cylinder wall varied in temperature with the steam inside, and in the case of an engine in which it did not so vary there could hardly be condensation as in this case. The Author had spoken of the mechanical efficiency of the engine as being rather low, and he had explained it by pointing to the large valves. These large valves no doubt took a good deal of driving, but on the other hand they were made

[1] Minutes of Proceedings Inst. C. E., vol. xcviii. p. 250.

large so as to have as small losses as possible from passage friction, and therefore there was probably a corresponding gain in the cost of the indicated power. If it were tried to obtain the last ounce of indicated work, it would very likely be got with a loss of real efficiency. The power lost in moving the valves, however, was not wholly lost, as it probably in great part reappeared as heat imparted to the steam. The Author had, he thought, lost sight of this in arriving at his heat-balance. In the case of trial 56 only 76.5 per cent. of the indicated power was converted into work, so that 23.5 per cent. was lost in friction. It was not unreasonable to suppose, he thought, that with these large valves the internal friction from them and from the glands and piston-rings accounted for a large portion of the total loss. Assuming that 15 per cent. of the work was consumed in internal friction in the case of trial 56, about 462 thermal units would have to be added to the radiation account, because the condensing water would in-

clude that amount of heat received from the indicated HP. wasted in friction, which heat would in Table I, line 35, be reckoned twice over unless 464 heat-units were transferred to the radiation account. With regard to all the methods of making the trials, they appeared to him to err on the side of complexity. From the fact, however, mentioned, that several trials could be made under similar conditions, the results not varying by more than 1 per cent., he concluded that the leak was a constant one, and this being so, it appeared extraordinary that it had not been located.

He thought, in conclusion, that it might be interesting to compare the Author's results with a condensing trial of one of his own engines.[1] In this trial, which was not conducted under conditions favorable to the highest economy, but only for comparison with a trial made with the same number of expansions exhausting into the atmosphere, the consumption of steam calculated as

[1] Minutes of Proceedings Inst. C. E., vol. xcvi. p. 255.

feed-water pumped into the boiler was 15.2 lbs. per indicated HP.-hour. The mean admission pressure was 168 lbs. absolute, and the back pressure in the condenser 4.4 lbs.; he was sorry that he had not the steam pressure in the high-pressure steam-chest for exact comparison with the Author's trials, but he would take it at 5 lbs. higher than the mean admission pressure in the cylinder. The higher and lower temperatures were then 369°.5 Fahrenheit and 156°.7. The water theoretically required per indicated HP.-hour between these limits was 10.26 lbs., and the water actually used being 15.2 lbs., the thermal efficiency was 67.5, or almost exactly the same figure as that found in the Author's best trial, if the calculation was made in the same manner. It would be observed, therefore, that the thermal efficiency of the Author's engine could hardly be said to be unprecedented, as even an unjacketed engine worked with an eight-fold expansion had given the same thermal efficiency, when calculated in exactly the same way, ex-

cept that the Author's figures were based on water coming out of the hot-well, and on jacket-water estimated rather than measured, while his own were for water pumped into the boiler.

The Author's figures for unjacketed trials were absolutely worse than his own, although the pressure used was greater and the vacuum better, being 17.4 pounds, 16.0 pounds, and 15.9 pounds for steam of 200 pounds pressure and calculated from the hot-well discharge, against 15.2 pounds for steam of 168 pounds pressure and calculated from the feed-water. The Owens College engine had been worked, so far as he could ascertain from the diagrams, with a ratio of expansion in the jacketed trials of 20 to 25, and in the unjacketed of something like 15. In his own trial, quoted above, the ratio of expansion was only a little over 8, which would give some idea of the relative sizes of the engine. In his own case, too, the power developed, 37.66 indicated HP., was much smaller than in the case of the Author's best trials, 72.1 indicated HP.

Mr. J. I. Thornycroft said he would confine his remarks particularly to the subject of the engine being divided. He thought it was an important thing in the engine, and he must say, though not in defense of it, that the engine being divided had not been sufficiently appreciated. It appeared to him that, for the purpose of the engine under discussion, it was a most suitable arrangement. Had the engine been constructed as some speakers recommended, it would have been made with a certain ratio of cylinders which could not be altered, and when it had been once tried and its performance ascertained, with the point of cut-off properly adjusted in the several cylinders, there would be little more to learn from the engine. But a compound engine, where the different parts could be run at different speeds, was one in which, if the engine were quite tight and perfect, the ratio of the cylinders could be altered, as far as possible; in fact, within almost inconceivable limits. He considered that in dividing the engine a

great field for inquiry had been opened, because it could be made to represent any particular ratio of cylinders which it was desired to examine. Although the brake had been criticised he thought it was an object of great interest. The Author had taken Mr. Froude's brake; but finding a defect in it, which at the time was not quite understood, he had turned that defect into a very important and valuable feature. The vortex in that brake was liable to become hollow, and by allowing the water to run out he could regulate the power of the brake within quite sufficient limits for all that was required. When the Author asserted that the engine could be run at from any speed down to 20 revolutions per minute with the load on the brake unchanged, so as to be independent of the speed of the engine, he thought it showed a great success, and the ability to make experiments with three independent engines running at the same time must be, to a certain extent, attributable to the success with which he encountered that

difficult problem, and made a brake which would answer his purpose. The great utility of the engine being divided was shown in the diagrams where the jacketed and unjacketed engines were compared. It was shown that if the engine was jacketed, to get the best results there must be a different ratio of cylinders. Mr. Bodmer had put forward a diagram (*Fig.* 5) showing that in the unjacketed engine he used a different ratio of cylinders, and that the jacketed engine was credited with more economy than it would be if it had a smaller ratio of cylinders. That argument might be turned the other way. It meant that had an unjacketed engine been obliged to run with as large a cylinder as a jacketed engine, the work in the unjacketed engine would have been worse than the Author had shown. He would ask the Author to be kind enough to give not only the good results, but the bad results he might get by using an improper ratio of cylinders for the particular cut-off, because it was not true

that a number of useful experiments could be tried with a given ratio of cylinders by varying the cut-off.

Mr. W. B. Bryan regretted that the Author had not measured the actual amount of feed-water at the trials, so as to compare it with the water discharged from the engines. In a triple-expansion engine which he was about to describe, he had found the missing quantity of water to which the Author referred to vary approximately from 5 to 10 per cent. In endeavoring to compare the work of a triple-expansion pumping-engine belonging to the East London Water-Works Company with that at the Owens College, a few words of explanation were necessary. In each case he had taken the net HP.; in the Owens College engine the brake HP., in the pumping-engine the work actually done in the pumps. The pumping-engine was of the inverted marine type, and the cylinders were 18 inches, 30½ inches, and 51 inches in diameter, with a 3-foot stroke. The high-pressure cylinder jacket was in circula-

tion with the boiler; the intermediate pressure and low-pressure cylinder jackets discharged their condensed water through steam-traps to waste. The normal speed was 23 to 24 revolutions per minute. The plunger-pumps, three in number, were worked direct off the crossheads. The cranks were set at 120°, and the crank-shaft was prolonged so as to actuate two 18-inch deep well pumps 190 feet below the surface, through two bell-cranks. There were eight bearings. The engine had a surface-condenser, the circulation water being that raised from the well, passing to the plunger-pumps. It would thus be seen that the engine had five pumps to actuate, besides a large number of working parts causing friction. The main steam-pipe was 75 feet long, and the steam-pressure 130 pounds per square inch. The conditions, as regarded the amount of friction, were in favor of the Owens College engine. A number of trials had been made during the past two years, and the example given showed

only average results. The feed-water, discharge from hot-well and jackets, were all from measurements in tanks, the tanks having been graduated from exact weights of water placed therein. All

PUMPING-ENGINE.

	Lbs.
Feed-water, per hour	2,225.0
Water discharged from the hot-well, per hour	1,937.5
" " " lp. cylinder jacket	77.1
" " " lp. " "	41.1
Total water discharged, per hour	2,055.7
Difference between feed-water and water discharged per hour	169 3
Water condensed in hp. jacket returned to boiler	110.0
Pump HP	140.0
Water per pump HP. discharged from hot-well per hour	13.84
Water per pump HP. passing through engine, including all jacket-water, per hour	15.47
Indicated HP.	160.0
Water per indicated HP. passing through engine, including all jacket-water, per hour	13.53

THE OWENS COLLEGE ENGINE.

Trials with cylinder jackets at boiler-pressure.

——	No. 44	No. 33	No. 56
	Lbs.	Lbs.	Lbs.
Water discharged from hot-well and jacket-water per hour	516.00	793.20	977.40
Brake HP	26.82	45.32	55.15
Water per brake HP. per hour	19.60	17.50	17.72
" per indicated HP. per hour	15.53	14.23	13.56

trials were made during the ordinary working of the engine. The power exerted was much less than the engine was designed for, on account of the water in the well having risen to such a height

as to drown the pumps to the extent of 130 feet. All the trials were of eight hours' duration. In the ordinary working of the pumping-engine, just described, the feed-water and the water required to make up the missing quantity were recorded daily. The pump HP. was taken from the water-pressure resistance in the plunger-pumps and lift in the well-pumps, there being no deduction or addition for any purpose. He might add that the efficiency of the engine in dividing the pump HP. by the indicated HP. for the whole of the trials worked out to between 87½ and 88 per cent. All these were calculated from diagrams taken from a day's running, the diagrams being recorded every quarter of an hour. He thought those results would show that the Author's engine must take the second place.

Mr. Charles E. Cowper thought the Author must be congratulated upon his excellent arrangement for making a variety of trials. After the first fifty-five trials it was found that some valve was leaking; he looked to the fifty-sixth trial

and hoped it would be better, but found it stated that even in this case steam was leaking into the receiver at the rate of one-half pound per minute, which was rather serious on a total consumption of 12 pounds. Such a result might afford a very excellent text for a professor to lecture his students upon, but he thought that experiments in which leaks had not taken such an important part would have been more valuable as well as more suitable to place before that institution. The Author spoke of the completeness of the system for checking results. Now what did he find in the paper? In No. 41 he found the radiation was 421 units; the Author said that was 500 units too small. There was an "error of observation somewhere." He thought it would have been better had the Author thrown aside the results of that trial. But there was a more serious objection, to which other speakers had alluded, namely, measuring the quantity from the hot-well, and not from the feed in the usual way. The Author stated that the feed was carefully

measured, with the result that it was from 5 to 10 per cent. greater than the discharge from the hot-well. That 5 to 10 per cent. unexplained loss seriously interfered with the results, and why the Author should charge that on the boiler he did not know. He thought that the boiler, tested before and after the trials, was more likely to be tight than the engine, which had twelve valves, nine piston and valve-rod stuffing boxes, seventy flanges and unions, and one hundred elbows, tees, etc.; altogether there were probably two or three hundred joints where leakage might occur. He would ask the Author whether he made independent boiler trials, and whether, in that case, he measured the water in the same way as in the engine trials, or whether he measured it from the feed-water, and missed out altogether the 5 or 10 per cent.? Mr. Cowper had found by experiment that still water at the temperature of the hot-well would evaporate at the rate of one pound per hour for each five square feet of surface. Of course, when

there were waves, the evaporation would be more, and in the present case there were two or three tanks, but he did not know what the surface amounted to. It had already been mentioned that the hot air would come out saturated with moisture from the air-pump, and would so carry away some of the water, as well as some of the heat. With regard to the difference in temperature between the hot-well and the feed-tank, he found that the Author had given it as in some cases zero, and in some between 20° and 30°, and he could only imagine that there was here again some error of observation. Mr. Cowper had selected from the Tables in the paper some of the most important figures (from a pure science point of view), and had drawn up a balance-sheet, placing on the debtor side all the heat in thermal units supplied to the engine, and showing on the creditor side the disposal of that heat, and had calculated the percentage in each case. He might say that he had taken those figures from Table I "without prejudice,"

that was to say, under protest, because he objected to the first basis of measurement from the hot-well. The result was 17.7 per cent. of indicated work, which was to be discounted by the 5 or 10 per cent. for error, which made 17 per cent.

BALANCE-SHEET—TRIAL NO. 56.

Dr.

Line.	——	T. U.	Per Cent.
(44)	Steam in cylinders.....	14,154	81
(47)	Jackets.....	3,325	19
(48)		17,479	100.0

Cr.

Line.	——	T. U.	Per Cent.
(34)	Indicated work.....	3,085	17.7
(33)	Rejected in condenser.....	12,862	73.6
(41)	Radiation.....	1,176	6.7
(40)	Loss from hot-well	356	2.0
(42)		17,479	100.0

or 16. It was still a very good result. He might mention that some large pumping engines, which were compound, not triple-expansion, gave a little over 15 per cent. indicated work.

Professor Osborne Reynolds in reply said he would first beg leave to repeat what was stated in the paper, that the purpose of the investigation was to determine the exact manner in which the separate parts or organs of the steam-engine performed their several functions. With this view each principal organ had been designed so that it might perform its own special function as perfectly as possible, and at the same time allow of its performance being measured irrespective of the action of the other organs. This necessitated the serious hampering of the engines, taken, as a whole, in any economic competitions; but this had been done without the smallest hesitation, any such competition being entirely outside the purpose for which they were intended. So, in making the trials, and reducing the results, every effort had been made to obtain and express the results of each organ separately. It was the purpose of a reduced diagram to show in detail how far the primary

organs of the engine, pistons, cylinders and valves, etc., had performed their part in rendering effective such steam-pressure as was possible under the conditions of working, and further, to separate as much as possible the sources of inefficiency in these organs. Until some such system as that adopted in this research became general, engine-trials would fail to serve what should be their most important purpose, that of showing the immediate sources of inefficiency. These diagrams did not seem to have been well understood by some of those who took part in the discussion, particularly by Mr. Willans, who attributed to a late cut-off what was really due to the setting back of the diagram, so that the ideal compression-line came to the point of zero volume, and appeared to think that such was the shape of the actual diagrams. As an illustration of the importance of such diagrams, in which the actual diagrams were set back by the ideal compression-curve, and then compared with the saturated curve, he might point

out that had Mr. Willans adopted this plan in his search for the missing quantity, he could not have fallen into the mistake which he had made in estimating the steam accounted for in the cylinder. By taking the apparent steam in the cylinder, less that in the clearance at the point of admission, as the steam accounted for, instead of the apparent steam, less that in the cylinder and clearance at the point of compression, he had taken all the steam condensed during compression as accounted for, and thereby consistently under-estimated the missing quantity, which was the main object of his search. In the high-pressure cylinders, where the compression reached nearly to the pressure of admission, the condensation during compression would be very considerable compared with the quantity which had been found missing at cut-off. Again, had Mr. Willans used the mean diagram to show the missing quantity, instead of trying to determine it for one point, and that the point of cut-off, it would have appeared that, in

the cases of the higher speed, at this point the missing quantity was apparently less on account of the inertia of the indicator-piston. He hoped Mr. Willans would now be induced to make another reduction of his indicator-diagrams by means of the mean diagram, after the method described. In this case he had no doubt Mr. Willans would find the missing quantity at cut-off in his high-pressure cylinder increased some 6 or 7 per cent. at the higher speeds, and that the cylinder-condensation varied with the square roots of the speed. The trials of Mr. Willans appeared to have been so carefully made, and the engines themselves so admirably adapted to test the effect of speed on cylinder-condensation, that it would be a great pity if the results, instead of furthering knowledge, were allowed to remain a stumbling-block for want of accurate reduction. Amongst others, Mr. Willans had asked how he explained the small effect of the jacket on the condensation in the high-pressure engine. In spite of the fact

that Mr. Willans stated that he had carefully read the paper, he must refer him to where he had carefully discussed this point, and given an explanation. The examination of the flow of heat through the walls of the cylinder, and the consequent influences on the efficiency of the steam-jackets of the thickness of the walls, and the state of steam as to excess of temperature and purity from admixture with air, was itself one of the objects he had primarily in view during the design and construction of the engine, and afforded a fair illustration of the character of the others. The accomplishment of this object, so far as the clear demonstration of the importance and character of these influences, was a result which, even if the research were carried no further, would constitute a sufficient scientific reward for the very considerable undertaking in instituting and working these engines. He hoped, however, to go further, both in extending the knowledge of these influences, and in clearing up other doubtful ques-

tions. The excess of heat in the discharged steam over that it possessed at release, was another important matter that had been examined and established in these trials. This point, however, had not been entered upon in the discussion. Mr. Beaumont's comments were merely an attempt to show that a discrepancy between the measured and theoretical heat discharged in one of the trials was owing to an error in estimating the heat theoretically discharged in all the trials without jackets. Mr. Beaumont overlooked the fact that by altering the formula to suit the one anomaly, he would bring an equal anomaly into both the others. In his formula Mr. Beaumont put a wrong construction on the total heat of evaporation, overlooking the fact that this included the external work done during evaporation against the pressure of the surrounding steam. The defect in the rider-valve mentioned, in connection with the discussion of any possible leakage that might explain the missing steam shown

in the diagrams, was only revealed by a most rigorous examination of the diagrams, and its effect on the distribution of steam and the efficiency of the diagrams was insensible, being only a slight wire-drawing for a little way from the cut-off. The leak through the steam-valve into the receiver No. II, in trial 56, was an incident of the trial, and only discovered in virtue of the exceptional facilities offered in these engines for observing the condition of each part separately. That these two slight defects should have been the only defects found in the six trials, notwithstanding the closeness of the watch kept on every part, was the very strongest assurance of not only the good, but the very high condition of the engines. Where no defects had been recorded, it might be that things had been perfect, but it was infinitely more probable that defects had not been looked for. In careful scientific work, it was a rule to record the largest errors, as affording definite information as to the degree of accuracy. This rule

he had followed. As to discarding any trial on account of these defects, which were revealed after or during the trials, this would seem very much like picking out favorable trials, to prevent which it was made a rule in this investigation to record every trial with its defects. The loss of steam or water was a defect common to all steam-engines and boilers, though the extent of this loss was seldom known, even approximately. Indeed, it was only in the case of surface-condensing engines that it could be measured at all, and then the apparent loss was liable to be less than the actual loss, on account of water leaking back into the condenser. It was almost impossible to determine the leakage of a surface condenser working under ordinary conditions. Like a boiler it might be tested cold and found tight, yet leak badly when hot. The Owens College engines afforded exceptional opportunities for testing the condenser. No water but what passed through the engines was admitted to the condenser; while,

not only was this water always measured, but also the heat given up by this water; and hence the heat showed how far the water entered as steam or leaked in as water. This check might not be within 3 or 4 per cent. of the water passing through the engines, so that with 20 lbs. a minute it would still be possible to overlook something like $\frac{1}{2}$ lb. per minute. It was, however, possible to keep the condenser under the same conditions as regarded pressure and temperature when passing 1 lb. per minute as when passing 20 lbs., and then the heat checked the water to $\frac{1}{20}$ lb. per minute. Several trials of this kind had been made, and had shown no determinable leak whatever. In this way the actual difference in the feed and the discharge from the hot-well was shown to represent, not only lost water, but all the water lost. As regarded the question how far 5 or 10 per cent. was a usual loss of water, it might be noticed that the statement of loss given in the paper referred only to triple-expansion trials with 200 lbs.

steam-pressure, conditions on the extreme limit of practice, so that there was every probability that the loss of water in these engines would be beyond the ordinary. He had heard from many engineers experienced in working marine engines that 5 per cent. was the minimum loss at high-pressure. Mr. Bryan stated that he had found the lost water from 5 to 10 per cent., and gave an instance in which the loss was 7.7 per cent. with about 130 lbs. of steam. At the time of writing the paper the loss occurring at lower pressures had not been determined; it had since been done, and it was found that up to a pressure of 90 lbs. the loss did not exceed 10 lbs. an hour, and hence the loss of from 40 to 50 lbs. an hour, which occurred at about 200 lbs. pressure, was to be attributed to the pressure. In the matter of liability to leakage from the engines and boiler, these engines were under much the same conditions as marine engines; but in respect of the water-heater and of the complicated system of pipes for supply-

ing the jackets, and under full pressure of steam, they were at a great disadvantage. On the other hand, very exceptional efforts had been made to render these engines tight. Every leak was stopped as soon as it was seen, and the boiler, economizer, and the entire jacket system were tested under cold-water pressure. On the last occasion when this was done, pressure was kept up at 200 lbs. per square inch for two hours with 20 lbs. of water. Nevertheless, during the engine-trial on the next day, 200 lbs. of water were lost in four hours. That the bulk of the lost water escaped as steam or water, through apertures in the envelope subject to high pressure and high temperature, seemed certain; but where these apertures were, whether numerous minute leaks or one larger leak, had not been ascertained. That no sensible quantity of water left the jacket-pipes was certain, these pipes being all well above the floor. If the pipes, therefore, leaked water, this water was evaporated off by the heat of the pipes, the

heat being replaced by the condensation of steam in the pipes, etc.; in this case, the quantity of water which returned from the jackets was not affected. The water that escaped from the glands of the high-pressure and the intermediate engines had been definitely shown to be very small. The glands had always been worked tight, namely, screwed up on the least suspicion of a leak. In some of the trials special precautions had been taken. In the trial already mentioned as following the cold-water test the glands were treated as usual, but the packing was rather old, so that when the 200 lbs. of water were lost, to be certain of the glands they were repacked, and another trial was made with the glands screwed and the rods kept cool by continuous tallowing. The glands were several times tested with cold polished steel, and no dew was found. The loss of water was 218 lbs. in the four hours. It had been suggested that the lost water had been discharged as vapor from the air-pump, or had been evaporated from

the tank. If this were so there would be the same loss in low-pressure trials as in high, for the condenser and tanks were under the same conditions. The only tank exposed was the feed-tank and the tip-can, in which the temperature of the water was within 2° of that in the hot-well. While considering that the feed-water could not contain more than one-hundredth of its volume of air at atmospheric pressure, the air discharged from the air-pump must be so small that it could not carry away any sensible moisture. He had, however, made arrangements to measure the air and vapor from the air-pump; these arrangements were not quite complete, but they enabled him to say that the volume of air and vapor did not appreciably exceed that of the one-twentieth part of the water discharged. He had replied at length to the remarks on this loss of water, because, for one reason, it seemed to have been the opinion that it had been rather slurred over. He hoped what he had now said would be sufficient

to show that, so far from such being the case, it had been pursued by every means at his disposal. Considering that such a loss of water was a matter of general occurrence with high pressures and surface-condensers, he considered its investigation to be of very great interest and importance. If the water was lost as steam, such a loss, if a necessary accompaniment of high pressures, must largely diminish the economical advantage, which would otherwise be realized. While the revelation, by the measurement of the feed and hot-well discharge, of a loss of more than 5 per cent. of the feed-water, which so far defied, not only all usual means of detection, but every means that he had been able to apply, must, until it was cleared up, cause grave doubts in accepting any conclusions as to the behavior of steam in the engine, based upon trials in which the feed alone had been taken as showing the quantity of steam passing through the engines.

The severe criticisms of Mr. Willans

seemed to demand some notice, which, however, had been made difficult by the attitude which he had taken. It was perfectly open to make any economic comparison so long as the results were dealt with accurately, and no misleading impression of them was conveyed. To begin with, Mr. Willans re-stated the Author's results, discounting the economic advantage (about 5 per cent.) which was obtained by means of the water separator returning the water condensed in the jackets to the boiler, measuring it on its way, which arrangement Mr. Willans in another place described as admirable. Not content with this, he attacked the method of estimating the jacket-water by measuring it accurately over an interval of five minutes every half-hour, because "only one-fifth of the jacket-water had in this trial gone through even a form of measurement." The water was collected in a carefully calibrated vessel, under full pressure of steam; while in place of this Mr. Willans proposed to take it out into the open air and weigh it, regard-

less of the fact that one-fifth part of the water would immediately pass off as steam. The measurement of jacket-water had always been a difficulty, and those having experience of this would, he felt sure, appreciate the advantage of this separator. There was no necessity to limit the intervals of measurement to five out of every thirty minutes; it had sometimes been measured for twenty-five out of the thirty minutes, yielding similar results for similar trials. In some trials all the water had been drawn off from the blow-off cock, keeping the water in the gauge at a certain level, so as not to allow steam to pass, the water being caught in buckets and weighed, a weighed quantity of cold water having first been put in the buckets to condense the steam. This was an accurate, though laborious method, which could not be used without the water-gauge. In this way both the gauge and the method had been severely checked. If the working of engines was so regular that indications for a period of less than half a

second every half hour were deemed sufficient, surely the definite measurement of the jacket-water for periods of five minutes every half hour might pass without question. Mr. Willans attempted to discredit the usual method of reducing diagrams by means of ten breadths. After asserting that "the measurement could only be accurately done by means of a planimeter," Mr. Willans proceeded to say that if the area of the low-pressure diagrams were taken by ten breadths, it would be at least 5 per cent. greater than if taken by the planimeter. It was difficult to believe that this was merely a guess; yet, if Mr. Willans had actually compared the measurements, he must have effected observation errors of at least 3.8 per cent. The figure was a sufficiently definite one, and as the essential error of the method of ten breadths depended solely on the figure, and not on the scale, and the area of a similar figure on a sufficiently large scale could be definitely and accurately measured, say to 0.02 per cent., and also

by ten breadths to the same degree of accuracy, the error of the method of ten breadths, as applied to this figure, admitted of definite determination. Having in his possession the original mean diagram, of which the scales were ten pounds to an inch and two cubic feet to an inch, from which the diagram in question was a copy, reduced first by the pentagraph and then by the engraver, he had the large diagram measured, so as to obtain its mean breadths to less than one-thousandth part, as well as by the method of ten breadths. He thus found that the excess by the method of ten breadths was 1.2 per cent. It might seem that these were very small errors to dispute about, but they were very important. If the ten-breadths method gave ordinary diagrams 5 or even 3 per cent. too large, it would be unfit for use, and all the results hitherto obtained by its aid would be subject to this error. Its standing error was, however, nothing approaching 5 per cent., not even 1 per cent.

Another respect, in which Mr. Willan's statements were calculated to cast discredit on the method of reducing the results of these trials, was that of the thermal and the theoretical efficiencies. In the paper, thermal efficiency had been used in a somewhat restricted sense, compared with the somewhat loose manner in which it was often used. Its application was restricted to the special organs in which the steam operated to perform work; thus it was used to express the ratio which the heat converted into work bore to the heat received into the engine proper, instead of confusing this heat with that lost by radiation, or otherwise lost. This restriction was defined in the paper and in Table II, "Thermal efficiency as given by heat discharged in condensing water." The purpose of so restricting it was obvious, namely, to keep the results depending on the action of these organs separate from other and quite independent organs. Mr. Willans objected to this step towards localizing the imperfections of

the engine, apparently because it had not been previously taken. He insisted on again mixing up the radiation with the internal action of the engines, as if it were wrong to separate them. This, however, was not so much the point. The restricted thermal efficiency depended on two distinct actions, one the action of the steam under the conditions aimed at in the engines, the other the action of the engine in fulfilling these conditions. Thus the thermal efficiency was the product of two efficiencies, one called by Rankine the efficiency of the steam, and commonly "theoretical efficiency," the other as yet without a name, but determined by the ratio which the thermal efficiency bore to the efficiency of the steam, and expressed as "percentage of theoretical efficiency." It was, if strictly determined, the most important quantity of all the results of the trials, as it expressed the efficiency of the primary organs of the engine in performing their functions, while, if incorrectly determined, it was simply a source

of confusion. This might be called the physical efficiency of the engine. To obtain correctly this physical efficiency, it was necessary to find the thermal efficiency, free from all confusion by extraneous circumstance, and also the efficiency of the steam according to the conditions under which it was used. In these trials with jackets the conditions aimed at were clear. Every effort had been made to keep the steam saturated while expanding from the pressure of admission to that at which it was released, to be discharged from the cylinder against the pressure of the condenser, the release-pressure being controlled by the load put upon the engine. Now the efficiency of dry saturated steam under such conditions had been known for more than thirty years, and there were no two opinions about it. Rankine and Clausius have given formulas which led to the same results. These theoretical efficiencies of the steam (Table III) had been obtained directly by interpolation in Rankine's tables, according to Ran-

kine's formula, given in Article 287 of his "Manual of the Steam-engine and other Prime Movers."

$$\text{Efficiency } \frac{U'}{h} = \frac{U_1 - U_2 + v_2 (p_2 - p_3)}{U_1 - U_2 + H_2 - h_4}$$

Mr. Willans, entirely ignoring the scientific importance of these terms, and observing only that, by taking the efficiency of steam under conditions which were not those aimed at in the trials, he could lower the percentage of theoretical efficiency, objected to the Author taking the correct efficiency for steam, and constructed a table in which he placed the theoretical efficiencies for the six trials as given in the paper, and above these, three rows of theoretical efficiencies, all decidedly higher than the Author's and differing from each other in an ascending scale, as if to prove, by mere weight of evidence, that the efficiencies given in Table III were too low.

The first two of these rows of efficiencies, "the theoretical efficiency of

Carnot's cycle," and "the theoretical efficiency of Clausius' perfect engine," were clearly inapplicable to the trials in which the conditions did not approximate to Carnot's cycle any more than they did to those of Clausius' perfect engine. The third row of efficiencies was headed: Theoretical Efficiency of Jacketed Engines: Heat utilized per pound of steam $= 1{,}438 \log._e \frac{A}{B} - 0.7$ (A — B). Standing thus directly over the results which the Author had given, obtained from Rankine's formula, there could have been but one intention, namely, to convey the idea that the Author had misused Rankine's formula and underestimated these theoretical efficiencies. What then was the fact? Mr. Willans had given a formula, emphatically not Rankine's, had headed the results calculated by this formula "Percentage of Theoretical Efficiency of Rankine's Jacketed Engine," and placed them immediately over the "Author's percentage of theoretical efficiency" (lines

W and X). Rankine had given no rule for the theoretical efficiency of a jacketed engine, but he gave a formula for the efficiency of dry saturated steam, which was the same as was now commonly meant by the theoretical efficiency of the jacketed engine. He also had given a formula for the heat utilized per pound of dry saturated steam, which was—

$$U' = a \log_e \frac{\tau_1}{\tau_2} - b(\tau_1 - \tau_3) + v(p_2 - p_3),$$

in which the suffix $_1$ referred to admission, $_2$ to release, and $_3$ to the condenser. It was this formula that Mr. Willans had used in concocting his own; he had put A for T_1, B for T_2, and left out the important final term. Nor was this all: in calculating the results Mr. Willans had, for B or T_a, the temperature of release, substituted the temperature of the condenser, which in these trials was some 60° Fahrenheit less than the temperature of release.

TABLE.

Number of the trial.....	44	33	56	41	35	40
Lbs. of coal put upon the fire during the trials, including 14 lbs. of wood taken as 7 lbs. of coal.....	321	487	411	365	521	509
Lbs. of coal deducted for fuel left unconsumed at the end of the trials.	21	21	21	21	21	21
Lbs. of coal consumed during the trials.....	300	466	390	344	500	488
Lbs. of water discharged from the hot-well supplied to the boiler at the temperature of the feed during the trials..	2,095	3,595	3,000	2,790	4,266	4,308
Lbs. of water added at the temperature of the feed to make up the feed during the trials.	..	288	..	..	278	273
Lbs. of jacket and priming water returned from the water separator by gravitation to the boiler at the temperature of the boiler during the trials, being the product of the rate per minute given in Table I, line 46, and the duration of the trial in minutes.....	944	1,365	960	595	725	646
Lbs. of total water supplied to the boiler during the trials, including water lost.....	..	5,248	..	..	5,239	5,227
Lbs. of water supplied to the boiler during the trials, excluding water lost.....	3,039	4,960	3,960	3,385	4,991	4,954

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